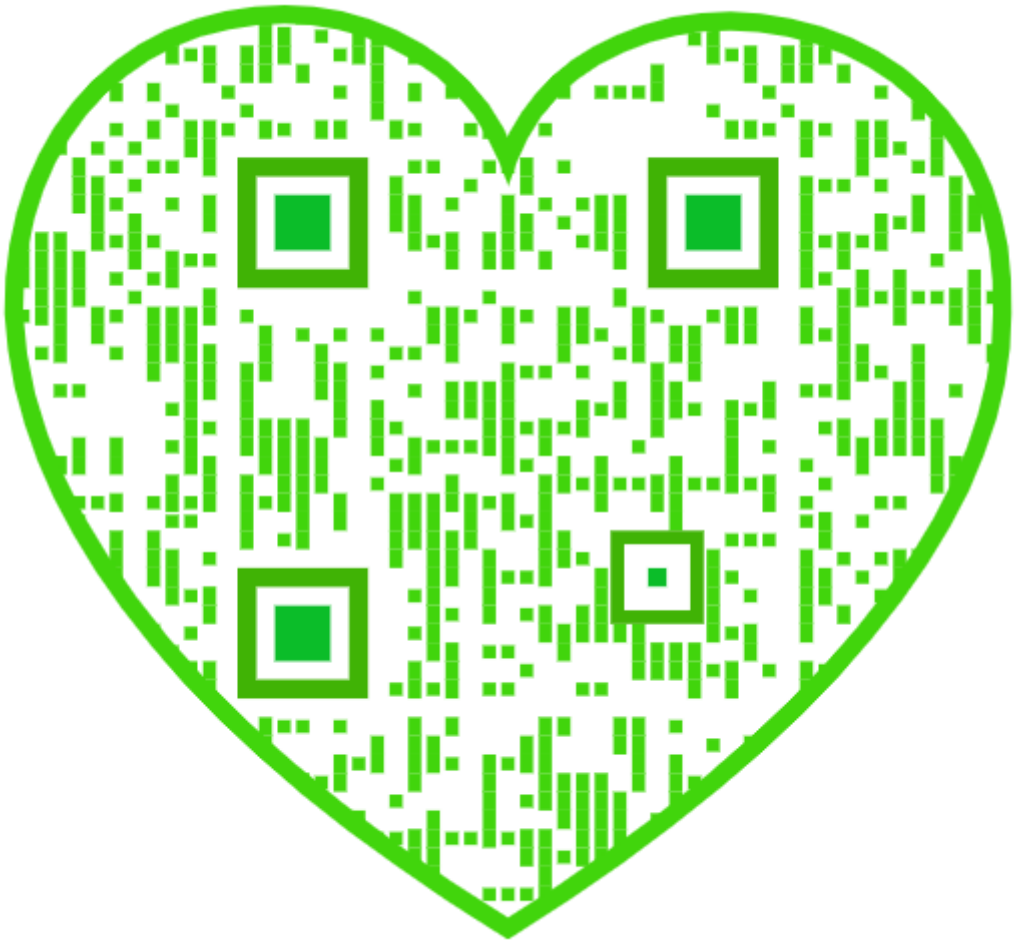

SEATON & ROUNTHWAITE'S

MARINE ENGINEERING POCKET-BOOK

A POCKET-BOOK OF
MARINE ENGINEERING
RULES AND TABLES.

FOR THE USE OF

MARINE ENGINEERS, NAVAL ARCHITECTS,
DESIGNERS, DRAUGHTSMEN,
SUPERINTENDENTS,

AND ALL ENGAGED IN THE DESIGN, CONSTRUCTION, AND CARE OF

MARINE MACHINERY, NAVAL & MERCANTILE.

BY

A. E. SEATON,
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FIFTEENTH EDITION, REVISED.

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PREFACE TO FIRST EDITION.

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A special Pocket-book of Memoranda, Tables, &c., has long been a desideratum with Marine Engineers. In the existing pocket-books, Marine Engineering matters are only dealt with generally, and such information as is given is in some cases very restricted, in others obsolete, and in all too scattered to be useful. We, ourselves, have experienced this want, and have heard on all hands the desire expressed for a Pocket-book in which Marine Engineering questions are dealt with thoroughly, are easy to find, and not "mixed up" with general information in such a way as to render the seeking of them difficult and tedious.

We therefore trust that in presenting this book to the public we have not only fulfilled the task we set ourselves, but have supplied this long-felt want in a manner that will prove satisfactory to all engaged in Marine Engineering affairs. While we have been careful to make the book of special value to Marine Engineers, we have omitted nothing, so far as we know, that would be of use and importance to others having to do with Ships and their machinery; at the same time, we have avoided the introduction of extraneous matter of only general interest, which would make the volume so bulky, and the arrangement of it so complex, as to very materially detract from its usefulness. Hence, we have, while not altogether neglecting past experience, but omitting information now almost only historic, devoted our attention generally to the most modern and approved practice.

We have dealt with steel as the material in general use, and not, as heretofore, an exceptional thing to be found only in high-class structures; the Tables of Weights, &c., are, therefore, given fully for this material.

Inasmuch as the practice in a considerable part of the Mercantile Marine is now more nearly approaching that followed in Naval ships, as to speed and economy of weight, than was formerly the case, the information and formulæ pertaining to light fast-running machinery have been elaborated and based on the most recent practice of the leading firms of Manufacturing Engineers.

In conclusion, we trust that the book may be received favourably, and found of use by practical men, and that any shortcomings may be overlooked on the score that it is the production of the spare moments of busy men, rather than of those having ample leisure.

A. E. S.
H. M. R.

October 1893.

PREFACE TO THE FIFTEENTH EDITION.

THIS is the Fifteenth Edition of the Pocket-book which was projected just thirty years ago, so that on the average a new edition has been published at intervals of two years. Each of these editions, by amendments and additions, has been brought thereby up to date, so as to accord with the best practice of the day as quickly as possible. This process is always a somewhat troublesome task for the Author and a more or less costly one for the Publishers; it would have been at one time much easier for both to merely reprint some more copies as the others were sold out. It has been, however, a satisfaction and reward to both Publisher and Author to find their enterprise and efforts rewarded by such a continued demand as to necessitate so many editions.

This new edition, however, has required a more drastic and extensive treatment than usual, inasmuch as so many of the old rules of the Board of Trade, of Lloyd's Register, of the British Corporation, and of the Bureau Veritas have been superseded by the new unified rules based on the recommendations of the British Marine Engineering Design and Construction Committee, and now adopted by these four authorities. This change has also necessitated a recasting of the tables and schedules based on these new arrangements.

The rules formerly in force in Government shipping circles in Germany have been expunged from this edition, as apparently they are no longer in general use there. Moreover, there is reason to anticipate that the recommendations of the British Marine Engineering Design and Construction Committee will be taken as the basis for those to be formulated there shortly.

The rules and regulations laid down for the guidance of the responsible marine surveyors in the United States of America are now given in a somewhat condensed form in the Appendix.

They differ considerably from those obtaining in this country by giving more permissive terms, and leaving much to the judgment of the designer and manufacturer. In passing it may be perhaps not out of place to note that much is likewise thereby added to their responsibilities.

Additions to the text in various parts have been made as a consequence of the extended experience with modern boilers, as also with machinery of every kind. It may be noted that while geared turbines and internal combustion engines of various sorts and designs are very much to the front now and continue in considerable demand, the triple and quadruple compound reciprocators continue, notwithstanding, to be held in high esteem by those who are of opinion that rate of consumption of fuel is not the only criterion of value. For the ordinary tramp cargo ship these steam engines are considered to be the most appropriate, as with them any fuel that will raise steam can be used, and besides their auxiliary machinery is such as to require small space and not much attention when at work. For the passenger ship oil fuel is, of course, the best in every way, and for steam raising it may be of a quality and cost very different from what the internal combustion engine requires for continued good working.

The saving in space occupied by the machinery, and the reduction in the engineers' crew make the internal combustion engine so attractive a proposition that most builders of marine machinery have now placed themselves in a position to supply them. It is to be regretted, however, that they mostly seem unable or unwilling to do so without making use of the patents and designs of foreign inventors. It is to be hoped, therefore, that those who have been content to develop those of British origin will meet with the success that their enterprise deserves.

The opposed piston oil engines and the double-acting oil engines are now being experimented with, and it will be interesting to follow their course, as it has been in the past to follow that of the various types of steam engine from the time when the S.S. *Comet* made a start over a hundred years ago.

A. E. S.

January 1922.

GENERAL TABLE OF CONTENTS.

PAGES

Prime Movers on Shipboard.—Steam-driven Reciprocators, Internal Combustion Reciprocators: Turbines: Combinations: Geared Turbines: Multiple Screws, . 1-2

Engine Power Measurements.—Nominal Horse-power, various Rules for: Estimated Horse-power: Normal I.H.P.: Standard sizes of Cylinders of N.E. Coast Makers: Board of Trade and Lloyd's Rules for N.H.P.: Indicated Horse-power, Shaft Horse-power, Nett Horse-power, Thrust Horse-power, &c.: Mean Pressure in Cylinder and Effective Pressure on Piston as shown by Indicators: Torsion Meters and Application of for Shaft Horse-power, ... 3-17

Efficiency of Marine Machinery.—Boiler, Steam, and Thermal Efficiencies: Mechanical and General Efficiency: Propulsive Efficiency: Froude's Curves: Friction: Experiments on Friction by various Scientists: Maximum output of Energy from a pound of Steam: Effect of Jacketing on Efficiency, ... 17-27

Propulsion of Ships and Resistance.—General Resistance, Residual Resistance: Admiralty Formulæ, derivation of them: Sir William White's Observations on Speed and Power: Cruiser and Destroyer Tests at varying Speeds: Results of Steamship Trials at ordinary and at very high Speeds by Reciprocators and Turbines: Kirk's Method of Analysis of Form: Table of Appropriate Angles of Entrance: Wetted Skin, Mumford's, Kirk's, and Seaton's Rules for: Seaton's Rules for Co-efficients of Form and Speed Factors: Curves of Power, &c.: Model Experiments: Co-efficients of Fineness: Friction of Immersed Surfaces: John's Co-efficients for computing Horse-power: True Mean Speed: Relation of Speeds and Powers: Tables of Times and Speeds: Table of Two-thirds Powers of Numbers from 100 to 56,000, . . 27-56

Compound Engines.—Multiple and Simple: Advantages of Triple over Simple Compound in Loads and Consumption of Steam: Cylinder Ratios in various Compound Engines: Mercantile and Normal Cylinder arrangements, 56-65

PAGES

Steam Expanding and doing Work.—Mean Pressure, equivalent in Compound Engines: Mean Pressure, Rules for: Steam Expansion, Isothermic and Adiabatic: Maximum Work: Table of Steam used Expansively: Effect of Clearance, Compression: Tables of Mean Pressure, taking account of Clearance; Mean Pressure expanded Adiabatically: Ratio of Mean Pressure in Practice to the Theoretical: Trials of Marine Engines, showing Mean Pressures, . 65–76

Piston Speeds and Revolutions of Engines.—As in Practice in the Navy and Mercantile Marine with various kinds of Engines: Rules for rate of Revolutions, Stroke of Piston, &c., ... 76–78

Cylinders.—Diameter of: Sizes of Ports, &c.: Flow of Steam: Ratios of Pipes and Passages: Strengths of: Thickness of Barrels and Liners, Steel and Cast Iron: Cylinder Ends and Covers: Cylinder Valve Boxes, &c.: Safety and Escape Valves: Drain Cocks: Starting and Auxiliary Valves: Column Feet and Bolts: Horizontal Engines: Oscillating Cylinders: Clearance of Pistons: Stuffing-boxes and Glands: Studs and Bolts, . . 78–105

Pistons.—Cast Iron and Cast Steel: Proportions and Scantlings: Forged Steel: Fittings and Details: Junk Rings and Packings, ... 106–113

Piston Rods.—Loads and Stresses: Proportions: Naval and Mercantile Practice: Fitting to Pistons: Guides and Guide Blocks: Fittings, &c., ... 113–116

Connecting-Rods.—Loads and Stresses: Proportions: Gudgeons: Brasses: Caps and Bolts: Scantlings of Bearings and Brasses, ... 116–122

Shafting.—Loads and Stresses: Bending and Twisting Moments: Torsional Stiffness: Solid and Hollow Shafts: Shafts of Screw Engines and Paddle Engines: Crankshafts: Equivalent Twisting Moments: Curves of Inertia Forces, Effect of on Shafts: Crank-pins and Main Bearings, Surfaces: Multiple Cranks, Effect of: Crank-arms: Shaft-coupling Bolts: Built-up Crankshafts: Crankshafts of Paddle Engines: Keys on Shafts: Board of Trade Rules for Shafts: Lloyd's Rules, . . 122–139

Thrust Shafts and Blocks.—Loads and Stresses: Indicated Thrust: Effective Horse-power producing Thrust: Pressure on Collars: Scantlings: Types of Blocks, . 139–142

Stern-Tubes.—How Made and Fitted: Proportions and Scantlings: Stern-bushes: Length of Bearing: Stuffing-box, &c., ... 142–145

Main Bearings of Crankshafts.—Arrangements of: Caps and Bolts: Loads and Stresses: Brasses, Types and Fitting: Materials: Engine Frames, Design, &c.: Columns, Loads and Stresses, ... 145–149

PAGES

Condensers.—Capacity of Jet Condensers: Injection Water: Low-pressure Steam, Properties of: Surface Condensers: Effect of Vacuum on Consumption: Forms and Arrangements, Weir's, Morison's: Cooling Surface: Temperature of Sea-water in various parts of the World: Cooling Water, Quantity and Application: Condenser Tubes, Sizes and Fitting of: Tube-plates: Number Tubes per square foot: Details and Fittings, . . . 149–158

Air Pumps.—Types of: Method of Working: Size of: Weir's Dual Pumps: Vacuum Augmenters: Rods, Bolts, &c.: Loads and Stresses: Pump Barrels, &c.: Scantlings of: Valves, &c.: Speed of Buckets: Suction and Discharge Pipes: Air Escapes, ... 158–163

Cooling Water Pumps.—Types of: Methods of Working: Capacity: Reciprocating Pumps, Details of: Valves, Rods, Pipes, &c.: Centrifugal Pumps: Sizes of Cylinders, Impellers, Pipes &c., ... 163–168

Feed and other Pumps.—Gross and Nett Feed-water: Capacity of Pumps: Valves, Rods, &c.: Suction and Delivery Pipes: Feed Tanks: Reserve Tanks: Feed Heaters: Board of Trade Rules: Lloyd's Rules for Feed Pumps, ... 168–172

Bilge Pumps, Pipes, and Fittings.—Capacity of: Boxes, Strainers, &c.: Directing Boxes: Board of Trade Regulations: Lloyd's Rules, ... 172–175

Pump Levers and Gear.—Arrangements: Sizes: Loads and Stresses: Links, Pins, and Crossheads, Surfaces and Sizes of: Details and Fittings, ... 175–177

Slide Valves for Steam Distribution, &c.—Travel: Single and Double Ports: Surface for Rubbing: Relief Rings: Port Openings and Leads: Valve Proportions: Trick Valves: Piston Path Diagram: Zeuner's Diagram for Common Valve Motion: Diagram showing Effect of "Notching up" (*a*) "Open" Rods, (*b*) "Crossed" Rods: Obliquity of Eccentric Rods: Diagram of Oscillating Cylinder Valve Motion, ... 177–186

Valve Gears.—Loads and Stresses: Valve Rods, Rules for: Guides: Links of various Kinds, and their Scantlings: Position of Suspension Pins: Proportions of Double-bar Links: Eccentrics, Construction and Scantlings: Straps and Rods: Joy's Valve Gear, Diagram of, . . 186–198

Reversing Gears for Valve Motions.—Types compared: Direct and All-round Gears: Weigh Shafts, sizes of: Steam Cylinders for: Worm Wheels of All-round Gears: 198–200

Steam Turning Gears.—Speed of: Steam Cylinders for: Worms and Wheels: Construction and Scantlings, . 200–202

Screw Propellers.—Numbers of Screws: Numbers of Blades to each: Shape of Blades: Section of Blades: Materials

PAGE

of Screws: Dimensions of Screws: Diameter: Pitch Ratio: Surface of Blade: Ratio of: Thrust: Slip, Real and Apparent: Rules for Diameter Pitch, Surface, &c.: Acting Surface: Scantlings: Materials: Attachment of Blades: Weight: Bosses, Studs, &c.: Particulars of various kinds of Screws, ... 200–214

Paddle-Wheel Propellers.—Common Radial: Effective Diameter: Thrust: Area of Floats: Slip, Apparent and Real: Number of Floats: Scantlings of Floats: Design of Wheel: Wheel Frames, Scantlings of: Shaft Bearings and Feathering Gear, ... 214–222

Sea Valves for Water Supply, &c.—Naval and Mercantile: Fittings and Materials, ... 222

Steam Turbines.—Types used for Propulsion of Ships: Reversing: Efficiency: Arrangements: Simple: Compound, Division over two or more Shafts: Geared: Hydraulic Transmission: Electric Transmission: Methods followed in Naval and Mercantile Ships: Combination with Reciprocators, R.M.S. *Olympic*, S.S. *Otaki*: Geared Turbines in S.S. *Vespasian*, S.S. *Normania*: Comparison of two Twin-Screw Reciprocating Engine Ships with S.S. *Sarnia*, ordinary Turbine, and S.S. *Normania*, Geared Turbine driven: Proportions of Screw Propellers for Turbine-driven Ships: Pressures on Journal of Turbine Shafts: Diameter of Shafting, Board of Trade and Bureau Veritas Rules for: Rotor Drums: Rate of Revolution: Blades of Turbines: Area through Blades: Diameter of Rotors: Exhaust Passages: Leakages: Water Tests, Admiralty, Board of Trade, &c.: Weight of Installations: Steam Consumption, Power developed, how measured: Torsion Meters: Shaft Horsepower: Consumption of S.S. *Lusitania* at various Speeds, also of S.S. *Otaki*, H.M.S. *Amethyst*: Trials of *Sarnia*, *Normania*, H.M. Cruisers "City" Class, S.S. *Reina Victoria Eugenia*: *Cairncross* and *Cairngowan* compared, ... 222–238

Internal Combustion Engines.—Various kinds: Gas, Petrol, Paraffin, and Heavy Oils: Oil Fuels used in such Engines: Values: Oil Engines Classified: Diesel and Semi-Diesel Engines: Two- and Four-stroke Cycles: Double-acting Oil Engine: Number of Cylinders: Reversing of Propeller: General Design: Cylinders: Pistons: Guides: Size of Shafting: Lloyd's and Bureau Veritas Rules for Shafts: Auxiliaries necessary for Oil Engines: Fuel Consumption of Diesels: Consumption at various Powers: Trials of various Oil Fuels: Efficiency, Mechanical, Thermal, and General: Mean Pressure: Rate of Revolution: Burmeister & Wain's Practice: Augsburg-

PAGES

Nürnberg Co.'s Practice: Sulzer's Practice: British Practice: Weights of Oil Engines, Space occupied by: Advantages of Diesel's: Indicated Horse-power Formulæ: Cylinder, Thickness of: Trials of Diesel Engine at Full and Slow Speed: Trials of S.S. *Eavestone*, *Selandia*, &c., &c., ... 238–252

Motor Boats, &c., using Petrol.—Board of Trade Rules for: Lloyd's Rules for Petrol and other Oils on Shipboard, ... 252–258

Superheated Steam.—Modern Practice: Economy: Maximum safe Temperature: Specific Heat of Superheated Steam: Total Heat of: Transmission of Superheated Steam: Maximum Work of: Heating Surface required for Superheating, ... 258–262

Skin Fitting and Valves.—Blow-off Valves: Method Attachment to Skin, Naval and Mercantile: Discharge Valves: Board of Trade Regulations: Lloyd's Rules: Details and Fittings, ... 262–265

Results of Trials of Engines.—Three-crank Triples: Four-crank Triples: Four-crank Quadruples, ... 266–268

Wire Gauges.—Various and their Equivalents, . . 270–271

Copper Pipes.—Suitable to various Conditions, . . 272–273

Wrought-Iron Pipes.—Suitable to various Conditions, . 274

Copper Pipe Flanges and Fittings.—Scantlings and Proportions, ... 275–276

Bronze and Cast-Steel Pipes.—Theoretical and Practical, . 277

Pipes in General.—Board of Trade Rules: Lloyd's Rules: Expansion by Temperature: Safety Devices: Steel Steam Pipes: Thickness of Steel Pipes in Practice: Solid-drawn Feed and Steam Pipes: Welded Pipes: Exhaust Pipes: Bending of Solid-drawn Pipes: Flanges: Admiralty Tests of Solid-drawn Steel Pipes, also of Welded Pipes, ... 278–286

Stop and Regulating Valves.—Construction: Details and Fittings, ... 286–287

Balancing Engines.—Various Forces: Methods of Static Balance: Dynamite Balancing: Inertia Forces: Yarrow Schlick-Tweedy System of Balancing four Crank Engines, ... 287–295

Geometry of Balancing Engines, ... 295–298

Boilers.—Fuels, various Solid and Liquid, their Characteristics and Values: British Thermal Unit: Mechanical Equivalent of Heat: Specific Heat: Total Heat of Combustion: Air required for Combustion: Composition of Fuels, Liquid and Solid: Oil Fuels used in U.S.A. Navy: Viscosity of Oil Fuels: Lloyd's Rules for Stowing and Using Oil Fuels: Admiralty Conditions of Contract for Oil Fuels: Rates of Combustion in Practice, . . 298–309

PAGE

Boilers and their Fittings.—Efficiency of Grates: Chimney Draught: Flow in Funnels: "Head" required for Draught: Fuel Consumed: Size and Height of Funnels: Table of Funnel Capacities: Scantlings of Funnels and Riveting: Forced Draught: Naval: Howden's System: Results of Forced Draught: Air Pressure in Boiler Rooms: Rates of Combustion and Evaporation, Practical Examples of: Express Boilers: Water Consumption of H.M.S. *Diana*, R.M.S. *Lusitania*, and various other Ships: Domestic Uses, &c., ... 309–321

Boilers.—Various Types and Designs of: Tank and Water-Tube Boilers: Cylindrical, various Sorts: Gunboat, Locomotive, Double-ended Water-Tube Boilers, (*a*) Large Tube, (*b*) Small Tube: Yarrow, Babcock, Niclausse, Hohenstein, Miyabara, Mumford, White-Forster, Thornycroft, &c., &c.: Total Heating Surface and Weight of Boilers, ... 321–325

Boilers.—Efficiency of various Types: Examples of, . . 326

Boilers.—Evaporation, Heating Surface, &c.: Efficiency of: Materials of Construction: Condition of Heating Surfaces: Circulation of Water: Tubes, sizes of: Water per pound of Fuel: Equivalent Evaporation from and at 212° F.: Examples of various Tank Boilers, . . 327–332

Boilers, Proportions of.—Furnaces, Size and Number of: Total Heating Surface Required for Various Services: Examples of Latest Practice: Water Consumption for Several Departments on Various Services: Auxiliary Machinery and Domestic Demands: Steam Room Allowance: Water Spaces: Pitch of Tubes: Multipliers to find Equivalent Evaporation: Weights of Various Installations: Various Designs Compared for Weight, . . 332–339

Boilers of Steel, Construction of.—Admiralty Tests of Steel: Board of Trade Tests and Conditions for Steel Materials, Plates, Forgings, Angle and Plain Bars, Rivets, Forgings, Castings, Tubes, Solid and Lap-welded, &c.: Board of Trade Conditions for Construction: Cylindrical Shells: Riveting of Various Kinds: Differing Thickness for Varying Tensile Strengths: New Board of Trade and Lloyd's Rules for Shells, and Quality of Steel: Tests for Boiler Material required by Lloyd's: Working Pressures: Riveting: Flat Surfaces and Stays, Board of Trade Rules for, with Tables: Dished Receiver Ends, &c.: Flat Surfaces and Stays, Lloyd's Rules for: Plates in Compression, Board of Trade and other Rules for: Girders and Stays: Stays and Tubes, Admiralty, Board of Trade, and other Rules for: Tables of Surfaces: Furnaces of all Kinds, Board of Trade and other Rules for: Testing by Water, ... 340–376

PAGES

Evaporators.—Board of Trade Rules and Regulations, . 376

Boilers, Construction.—Lloyd's General Rules for, . . 379

Boilers, Construction.—Board of Trade Rules for Shell Joints: Examples of Various Kinds of Riveting, . . 380–389

Boiler Work.—Supervision of, as required by British Admiralty, and the Procedure in Manufacture: Treatment of Mild Steel by Heat: Pickling Processes, . 389–392

Boiler Mountings and Fittings.—Stop Valves: Steam passed through pipes of various sizes: Safety Valves, Rules of Board of Trade and B.M.E.D. & C. Committee, &c. &c.: Spiral Springs, Rules for: Feed Valves and Pipes: Blow-off and Scum Valves: Water Gauges: Weight of Water at Varying Temperatures: Circulating Apparatus: Board of Trade Regulations for Mountings, &c.: Lloyd's Rules for same, ... 392–408

Furnace Fittings.—Doors, Size and Construction: Fire-bars: Bridges, ... 408–409

Ladders and Platforms.—Proportions and Scantlings, . 410

Engine and Boiler Seatings.—Arrangement and Scantlings, Screw Engines, Paddle Engines and Thrust Blocks: Holding-down Bolts: Stays: Boiler Seatings: Methods of Securing Boilers: Lloyd's Rules for Seatings, Bearers, Beams, Bulkheads, Shaft Tunnels, ... 411–414

Lloyd's Rules.—For Valves in Bulkheads, Openings in Decks, Coamings to Hatchways: Deck Casings, . . . 414–417

Steam Trawlers.—Lloyd's Rules for Engine and Boiler Rooms, ... 416

Pumps for Bilges, &c.—Lloyd's Rules for Sluice Valves, Sounding Pipes, Suction Pipes, Bilge Injection, &c., . 417–419

Surveys of Machinery.—Lloyd's Regulations for, . . 419–421

Spare Gear.—Lloyd's Requirements, ... 421–422

Board of Trade General Rules and Regulations for Machinery Department.—Auxiliary Engine Suctions: Bilge Injections: Spare Tiller: Rudder Chains, &c.: Steering Engine Pipes: Steam Steering Engine Gearing: Fire-hoses and Fittings: Stand-pipes and Cocks: Chocking Boilers, ... 422–426

Spare Gear desired by Board of Trade.—Main Engines: for Distillers: Survey of Distillers, ... 426–428

Chains and Ropes.—Admiralty Tests: Lloyd's Tests: Weight of Breaking Strengths: Special Flexible Wire Ropes: Hemp Ropes: Bullivant's Special Products: Admiralty Tarred Cordage, ... 429–436

Strength of Materials.—Cast Iron of various kinds: Iron Mixtures: Admiralty Requirements of Cast Iron: Wrought Iron of various Kinds and Qualities: Cast Steel: Admiralty and Lloyd's Tests for Steel Castings: Board of Trade Tests for Steel and Malleable Cast Iron: Steel

PAGES

Bars and Plates: Admiralty, Lloyd's, and British Corporation Tests for Wrought Steel, . . . 436–447
Materials.—Copper, Admiralty Specification: Common Bronze or Gun-metal: Admiralty Bronze: Phosphor and other Special Bronzes: Brass and Various Yellow Metals: Aluminium and its Alloys, &c., ... 447–452
Composition, Properties, and Costs of Various Metals, . 453–457
Plates, Bars, Rolled Shafts, &c.—As manufactured in Great Britain, and the Extras chargeable and other Conditions, ... 458–461
Beams and Girders.—Effects of various Loads, . . 463–470
Test Pressures on Flat Surfaces (Maximum Fluid), . . 471–473
Effect of Temperature on Metals, ... 474–477
Weights of Materials, &c., ... 479–495
Weights, &c., of Machinery, ... 496–497
Water, Fresh and Salt.—Information on, . . . 498–502
Oils and Lubricants.—Viscosity: Specific Gravity: Flash-points of, ... 502–504
Friction.—Co-efficients of, &c., ... 504–505
Conductivity of Metals.—Thermal, Electric, and Acoustic, . 506
Fuel Consumptions.—Solid and Liquid, . . . 508–509
Thermometers.—Fahrenheit, Celsius, and Reaumur compared, ... 510–516
Steam, Saturated.—Properties of, ... 517–525
Knots, Miles, and Kilometres compared, . . . 527–530
Metrical and British Standard Measures compared, . 530–546
Circles.—Properties: Tables, ... 547–564
Spheres and Cones.—Properties: Tables, . . . 565–566
Square Cubes and Roots of Numbers, . . . 567–611
Fourth Powers of Numbers, ... 612–613
Hyperbolic Logarithms, ... 614–616
Nomenclature and Definitions, ... 617–618
British Corporation.—Rules and Regulations for Machinery, 621–640
Bureau Veritas.—Rules and Regulations for Machinery, . 641–666
U.S.A. Government.—Rules and Regulations for Machinery, 667–675
Electric Lighting, &c.—Lloyd's Rules and Regulations for, ... 672–677
Refrigerating Machinery.—Lloyd's Rules and Regulations for, ... 678–685
Steering Gear.—Lloyd's Rules and Regulations for, . 686–687
Lloyd's Instructions to Surveyors *re* **Tests, &c.,** . . 688–694
Lloyd's Rules for Diesel and other Oil Engines:—
(*a*) Nominal Horse Power; (*b*) Rules for the Construction and Survey of Diesel Engines and Auxiliaries, 695–704
Lloyd's Rules for Screw Shafts and Stern Tubes, . 705
Distances of Various Principal Ports, &c. . . 706–718
Standard Specification of North-East Coast Engineers for Triple Compound Engines, ... 719–724

PAGES

Russian Weights and Measures compared with British and Metrical, ... 725–726
Hydraulic and Steam Tests of the Admiralty and Register Societies, &c., ... 727–729
Melting-points of Various Metals, . . . 730
Lloyd's New Unified Rules for the Survey and Construction of Engines and Boilers of Steam Vessels, . 731–756
Weight of Metal Plates per Square Foot, . . 757

Index, ... 758–770

LIST OF TABLES.

No.	Subject Matter.	
I.	Sizes of cylinders, &c., corresponding N.H.P.,	
II.	Nominal horse-power of triple compound engines,	
III.	,, ,, of quadruple compound engines,	
IV.	Constant multipliers for I.H.P.,	
V.	Maximum work from one pound of steam expanding and exhausting to condenser,	
VI.	Maximum work from one pound of steam expanding and exhausting to atmosphere	
VII.	Maximum work from one pound of steam expanding and exhausting against pressure,	
VIII.	Relation of powers and speeds of ships,	
IX.	Results of trials of steamships,	
X.	,, ,, ,, of high speeds,	3
XI.	Angles of entrance suitable for various speeds of ships,	
XII.	I.H.P. per 100 feet of wetted skin of different speeds,	
XIII.	Prism co-efficients of ships suitable for speeds and lengths,	
XIV.	Values of co-efficients in Admiralty speed and power formula,	
XV.	Co-efficients (A. W. John's) for computing E.H.P. from skin areas,	
XVI.	Resistances of various surfaces to water passage,	
XVII.	Speeds from times on measured mile,	
XVIII.	Values of $\frac{V}{\sqrt{\text{displacement}^2}}$ or $\frac{V}{D^{\frac{2}{3}}}$ power,	
XIX.	Efficiency of steam at various pressures,	
XX.	Ratios of cylinders of engines on compound systems,	
XXI.	Ratios of mean to initial pressure steam used expansively,	
XXII.	Factors for mean pressure, taking *clearance* into account,	
XXIII.	Mean pressures of steam expanding adiabatically,	
XXIV.	,, ,, ,, as found in practice,	7
XXV.	Results of trials of marine engines (twin screw and single),	7
XXVI.	Rates of revolution of main engines in practice,	7
XXVII.	Relation of stroke and revolutions of cargo steamers,	7
XXVIII.	Piston stroke of different classes of ships,	7
XXIX.	Speed of flow of steam through ports, &c.,	8
XXX.	Ratio of ports and pipes of steam cylinders,	8
XXXI.	Thickness of cylinder barrels of triple compound engines,	8
XXXII.	Thickness of cylinder barrels of ordinary compound engines,	84
XXXIII.	Thickness of cylinder barrels of torpedo boats, &c.,	84
XXXIV.	Cylinder liners of cast iron,	85

No.	Subject Matter.	Page.
XXXV.	Cylinder liners of forged steel,	86
XXXVI.	,, ends of triple compound engines,	88
XXXVII.	,, covers, cast steel,	91
XXXVIII.	,, relief valves,	95
XXXIX.	,, drain valves,	96
XL.	Piston clearances (axial),	99
XLI.	Stuffing-boxes, ordinary,	100
XLII.	,, ,, for metallic packing,	102
XLIII.	Studs and bolts, strength of,	103
XLIV.	Nuts and bolts, dimensions of,	105
XLV.	Pistons of cast steel,	108
XLVI.	,, of forged steel for light engines,	109
XLVII.	,, of cast iron,	110
XLVIII.	Piston-rods, strength of,	118
XLIX.	Brasses for bearings and rods,	121
L.	Shafts for screw engines and paddle engines,	124
LI.	Working pressures on bearings, pins, &c.,	128
LII.	Values of F_1 for various reciprocating engines,	138
LIII.	Values of K for tail shafts	139
LIV.	Thrust-blocks, surfaces of,	142
LV.	Stern-tubes, bushes, &c.,	144
LVI.	Weight, volume, temperature of low-pressure steam,	150
LVII.	Effect of vacuum on steam consumption,	152
LVIII.	Temperature of sea-water at various parts of the world,	153
LIX.	Cooling surface, allowance of, per I.H.P.,	154
LX.	Cooling water, ratio of, to steam condensed,	155
LXI.	Surface condenser, size of,	157
LXII.	Air and circulating pumps,	163
LXIII.	Feed pipes, deliveries through at service speeds,	170
LXIV.	Pump links, strength of,	175
LXV.	Proportions of double-bar reversing links,	190
LXVI.	,, of eccentric sheaves and straps,	194
LXVII.	Thickness of propeller blades at root,	209
LXVIII.	,, ,, ,, at tip,	210
LXIX.	Particulars of screw propellers in various ships,	213
LXIXA.	,, ,, ,, of high-speed ships,	214
LXX.	Performances of certain channel steamers,	226
LXXI.	Oils and other liquid fuels for internal combustion engines,	240
LXXII.	Consumption of oil at different powers,	247
LXXIIA.	Various oils, properties of,	247
LXXIIB.	Results of trials of an oil engine with various fuels,	247
LXXIII.	Rates of revolution of Augsburg-Nürnberg oil engines,	248
LXXIIIA.	Comparison of steam consumption,	259
LXXIV.	Specific of heat of steam at various temperatures,	259
LXXV.	Maximum work possible for 1 lb. of superheated steam expanding to 1 lb. pressure,	261
LXXVA.	Weight of superheated steam passed through pipes,	261
LXXVI.	Result of trials of triple-expansion three- and four-crank engines, &c.,	266
LXXVIA.	,, ,, of quadruple-expansion engines,	268
LXXVII.	Legal standard wire gauge and metrical equivalents,	269
LXXVIII.	Wire gauges and their equivalents,	270
LXXIX.	Thickness of copper pipes (L.S.G.),	272
LXXX.	,, ,, ,, by B.M.E.D. & C. C. rules,	273
LXXXI.	,, of iron pipes, by B.M.E.D. & C. C. rules,	274
LXXXII.	Flanges for copper pipes,	275

No.	Subject Matter.	
LXXXIII.	Thickness of bronze and cast-steel pipes, T pieces, &c.,	
LXXXIV.	,, of cold solid-drawn steel steam and feed pipes in 64ths of an inch,	
LXXXV.	,, of welded or riveted steam pipes in 64ths of an inch,	
LXXXVI.	,, of solid-drawn or riveted exhaust pipes in 64ths of an inch,	
LXXXVII.	Composition and value of fuels,	
LXXXVIIA.	,, ,, of liquid fuels,	
LXXXVIIB.	Consumption of various liquid fuels per 24 hours,	
LXXXVIIC.	Composition, &c., of certain liquid fuels as in practice,	
LXXXVIID.	Oil fuels as used in U.S.A. Navy boilers,	
LXXXVIIE.	,, ,, viscosity of,	
LXXXVIII.	Capacity of funnels for quantities of fuel burnt per hour,	
LXXXVIIIA.	Pitch, &c., of riveting for funnels, casings, &c. (Admiralty),	
LXXXIX.	Results of trials of certain boilers at full powers,	
XC.	Rates of combustion and evaporation (tank boilers),	
XCI.	Express boilers, particulars of surface, weight, &c.,	
XCII. and XCIIA.	Water-consumption trials of H.M.S. *Diana* and S.S. *Lusitania*,	
XCIII.	Consumption of water and fuel on trials of various ships,	
XCIV.	Thermal efficiency of various boilers,	
XCV.	Particulars of destroyers built by Messrs Yarrow & Co.,	
XCVI.	Comparison of water tube boilers with others in various ships,	
XCVII.	Leading particulars of some boilers made in recent years,	
XCVIII.	Particulars of large modern cylindrical boilers, scantlings,	
XCIX.	Total heating surface per I.H.P. of various ships,	
XCIXA.	Allowance of steam room in boilers,	
C.	Multipliers for converting weight of water evaporated to the equivalent from and at 212° F.,	
CI.	Relative weights of various boiler installations,	
CIA.	Comparison of steel boilers of various designs, Board of Trade rules for 160 lbs.,	
CII.	Admiralty tensile tests for steel boiler materials,	
CIII.	Board of Trade tensile tests for steel boiler materials,	
CIV.	Relative thickness of boiler plates for different tensile strengths,	
CV.	Board of Trade constants for flat surfaces,	
CVI.	Pitch of stays and area of flat surfaces of combustion chambers (B.M.E.D. & C. Committee),	
CVII.	Pitch of stays supporting flat plates not exposed to flame (B.M.E.D. & C. Committee rule),	
CVIII.	Pitch of stays supporting flat plates when fitted with washers (B.M.E.D. & C. Committee rule),	
CIX.	Working pressure, boiler shells of 28 tons tensile steel,	
CIXA.	,, ,, ,, 32 ,, ,,	
CX.	Lloyd's old rules for flat surfaces,	
CXI.	Corrugated furnaces, working pressure in lbs. per sq. inch,	
CXII.	Surface of plate supported by one screwed stay (B M.E.D. & C. Committee rule),	

No.	Subject matter.	Page.
CXIII.	Surface of plate supported by one stay of 28 tons tensile steel, B.M.E.D. & C. Committee rule,	373
CXIV.	Tests (mechanical) of boiler materials,	386
CXV.	Joints of plates, their several kinds and strengths,	389
CXVI. and CXVII.	Quantity of steam passed through pipes,	393
CXVIII.	Safety-valve springs, sizes of, by Board of Trade rules,	397
CXIX.	Weight of pure water at different temperatures,	400
CXX.	Safety-valve areas for different pressures, Board of Trade,	404
CXXI.	Ladders and gratings, scantlings of,	410
CXXII.	Sizes of bilge suction pipes (Lloyd's rules),	419
CXXIII.	Admiralty tests, &c., of stud-link chain cables,	429
CXXIIIA.	,, ,, short-link chains,	429
CXXIV.	Lloyd's ,, ,, stud-link chain cables,	430
CXXV.	Admiralty flexible steel wire ropes,	431
CXXVI.	Breaking strength of steel wire hawsers (Lloyd's),	432
CXXVIA.	Special flexible steel wire ropes,	433
CXXVII.	Bullivant's steel wire ropes, galvanised,	434
CXXVIII.	Mild plough steel wire crane ropes (Black),	435
CXXIX.	Admiralty tarred hemp cordage,	436
CXXX.	Composition and qualities of cast iron,	437
CXXXI.	Comparative requirements for steel castings,	445
CXXXII.	Composition of white (bearing) metals,	453
CXXXIII.	Properties of various metals,	454
CXXXIV.	Prices of materials,	456
CXXXV.	Safe working stresses on various metals,	462
CXXXVI.	Bending moments, &c., of beams,	463
CXXXVII.	Moments of inertia, modulus, &c., of some sections,	467
CXXXVIII.	Forms of beams of uniform strength,	469
CXXXIX.	Greatest fluid test pressure on flat surfaces of cast iron,	471
CXL.	Greatest fluid test pressure on flat surfaces of steel and bronzes,	472
CXLI.	Greatest fluid test pressure on flat surfaces of bronze castings,	473
CXLII.	Expansion of metals, &c., for rises in temperature,	474
CXLIII.	Effect of temperature on certain metals,	475
CXLIV.	Melting-points of various metals,	476
CXLIVA.	Melting-points of various alloys,	476
CXLIVB.	Specific heat of various materials,	477
CXLIVC.	Thermal conductivity of metals,	477
CXLIVD.	Electrical resistance of metals,	478
CXLV.	Weights of various materials,	479
CXLVI.	Weight of round and square bars of wrought iron,	480
CXLVII.	,, ,, ,, mild steel,	481
CXLVIIA.	,, round steel shafts,	482
CXLVIIB.	,, ,, hollow steel shafts,	482
CXLVIII.	,, flat bars of wrought iron,	483
CXLIX.	,, ,, mild steel,	484
CXLIXA.	,, large rectangular section steel bars,	485
CL.	,, angle bars of wrought iron,	486
CLI.	,, ,, mild steel,	487
CLII.	,, boiler tubes of wrought iron,	488
CLIII.	Standard list of sizes and prices, &c., of welded boiler tubes,	490
CLIV.	Weight of large steel tubes,	491
CLV.	,, seamless copper tubes,	492

No.	Subject Matter.	Page.
CLVI.	Whitworth's standard gas threads,	494
CLVII.	Weight of brass condenser tubes,	494
CLVIII.	,, lead pipes,	495
CLIX.	,, sheet metals,	495
CLX.	,, engines and boiler installations of all kinds,	496
CLXI.	Surface of tubes in square feet,	498
CLXII.	Weight of fresh-water per volume,	498
CLXIII.	,, salt ,, ,,	499
CLXIV.	Composition of solid matter in feed waters,	499
CLXV.	Quantity of solid matter in sea-waters and various seas,	500
CLXVI.	Composition of solid matter in sea-waters,	500
CLXVII.	Weight of cubic foot of sea-water at various ports, &c.,	501
CLXVIII.	Boiling-points of sea-waters of different densities,	502
CLXIX.	Viscosity of oils at various temperatures,	502
CLXX.	Characteristics of various lubricating oils of good quality,	503
CLXXI.	Boiling, setting, and flash points of various oils, &c.,	504
CLXXII.	Co-efficients of friction of various substances,	504
CLXXIII.	Conductivity, acoustic, electrical, thermal, of metals,	506
CLXXIV.	Various gases, properties of,	506
CLXXV.	Pressure of water due to various "heads,"	507
CLXXVI.	Coal consumed per day at various rates of consumption,	508
CLXXVII.	Oil fuels ,, ,, ,, ,,	509
CLXXVIII.	Comparison of thermometers,	510
CLXXIX.	Properties of saturated steam, temperature, density, heat, &c.,	517
CLXXIXA.	Total heat of evaporation from and at differing temperatures,	525
CLXXX.	Knots, miles, kilometres, &c.,	527
CLXXXI.	Kilometres and Admiralty knots,	530
CLXXXII.	Millimetres and inches,	531
CLXXXIIA.	English feet and French metres,	536
CLXXXIII.	Decimal equivalents of fractions of an inch,	539
CLXXXIV.	Metrical ,, ,, ,,	539
CLXXXV.	Square feet and square metres,	540
CLXXXVI.	Square metres and square feet,	541
CLXXXVII.	English pounds avoirdupois and kilogrammes,	542
CLXXXVIIA.	Kilogrammes and pounds avoirdupois	543
CLXXXVIII.	Lbs. per sq. in. and kilogrammes per sq. centimetre,	544
CLXXXIX.	Kilogrammes per sq. centimetre and lbs. per sq. in.,	545
CXC.	Kilogrammes per square millimetre and tons per sq. in.	546
CXCI.	Areas of segments of circles,	547
CXCII.	,, of circles,	550
CXCIII.	Circumferences of circles,	557
CXCIV.	Areas and circumferences of small circles,	564
CXCV.	Spheres, volume and surface of, also that of cones,	565
CXCVI.	Squares, cubes, square roots, cube roots, and reciprocals,	567
CXCVII.	Fourth power of numbers,	612
CXCVIIA.	Fourth root of numbers,	612
CXCVIIB.	Fourth power of shaft diameters,	613
CXCVIII.	Tons of water delivered through pipes with loss of head =5 lbs.,	614
CXCIX.	Hyperbolic logarithms,	614
CXCIXA.	Power transmitted per revolution of shafts,	616
CC.	Horse power transmissible by shafts,	619
CCI.	Insulating materials for boilers,	620
CCII.	Refractory materials for furnaces	620
	Weight of metal plates per square foot	757

MARINE ENGINEERING RULES AND TABLES.

MARINE ENGINES—VARIOUS KINDS OF.

The prime movers employed on shipboard for driving the propellers are to-day:—

(1) Reciprocating Engines; (2) Turbines; (3) Combinations of both.

1. **Reciprocating Engines** are worked by means of steam or the combustion within their cylinders of gas or spray from liquid fuels, or the gas from solid fuel.

Steam-driven reciprocators are direct acting, inverted in the case of screw ships and inclined for paddle ships. They are invariably of one of the compound types—that is, the steam is expanded and acts in a series of cylinders, instead of in one only. Marine engines of all steamers, except very special ones, such as tugs, where economy of fuel is not of prime importance, are of the triple or quadruple expansion principle, so that there are three or more cylinders to each. Fast-running engines in express steamers and naval ships, whether triple or quadruple, have usually four cylinders operating on four cranks.

Oil-driven reciprocators.—There are two kinds, known generally as the *Diesel* and the *Semi-Diesel*; they both consume heavy oil, but differ fundamentally as to the degree of compression of the air-charge before the liquid fuel is injected. In the case of the Diesel, the air compression (usually 35 atmospheres at least) results in a temperature high enough to ignite the incoming oil spray. In the Semi-Diesel the compression is considerably less, and the temperature is insufficient for the purpose of ignition; in it hot bulb or plate igniters are fitted.

The Semi-Diesel engine works on the two-stroke cycle whereby an explosion occurs at each revolution.

The Diesel engine is sometimes worked on this cycle and sometimes on the four-stroke cycle, with an explosion only at each alternate revolution, that is, at one in four strokes.

In small craft, paraffin is sometimes used as fuel with engines working on the four-stroke cycle. Generally all these oil engines are single-acting, as double-acting ones have not so far proved to be satisfactory in continuous running.

2. **The Turbine** is a rotatory engine and essentially a **velocity** machine, as against the previous kind, which are all actuated by **pressure**. It derives its motion and power entirely from the kinetic energy **of the** steam particles, to which great velocity is imparted by the **expansion** during the lowering of pressure from the initial to that at exit **to the** condenser.

The expansion may take place in one stage, as in the De **Laval** turbine, or it may be in a series of stages, just as in **compound** reciprocators.

Other rotatory machines whose motion and power are due to **steam** pressure acting on pistons or their equivalents have been tried, **but in** small units only have they been successful.

3. The combination of a turbine with a reciprocator has been **carried** out on a very large scale in R.M.S. *Olympic*, and on other ocean-going steamships of various sizes, all with satisfactory results. **By** means of the low-pressure turbine the energy remaining in the **exhaust** steam from a triple compound reciprocator is fully utilised, and **the** very high vacuum so easily and cheaply maintained in a marine **condenser** thus made full use of effectively. The saving in fuel by this combination is generally about 15 per cent. over that of the triple or quadruple engine, or there is an increase in power developed **to a** corresponding extent from the same expenditure of fuel in each case.

Reciprocators are nearly always connected direct to the propeller shafting, as their rate of revolution is not unsuitable to that of the propellers.

All Turbines are now geared by pinion and wheels to the propeller shaft, so that they may not have so low a rate of revolution as that required for the high efficiency of a satisfactory screw. The efficiency of the turbine to be good requires high peripheral velocity or a large number of stages for expansion, the former being desirable.

Ungeared Turbines used to be divided so that small ships had three screws and large full-powered ones four screws. Just before gearing was adopted the twin screw arrangement, with a complete turbine combination to each, was preferred for "Destroyers" and "Flotilla Leaders."

Geared Turbines may be each divided into two parts, the high pressure and the low pressure. For cargo ships of low power one screw is usual, for higher powers multiple screws obtain, so that Naval ships of very high speed have as many as four screws. The gearing may be single for small powers, but now is usually double for larger powers, and is preferable.

Large ships with high power, even when driven by reciprocators, were sometimes designed with three screws both in the Naval and mercantile services, especially in some foreign ones.

ENGINE POWER—MEASUREMENT OF.

Nominal Horse-power, as understood by Watt, was a measure of the commercial value of an engine, being the power it might be expected to develop in ordinary work. Gradually, however, as it became possible to construct boilers to supply steam at higher than atmospheric pressure, the powers developed by engines exceeded the nominal horse-powers,

until Watt's rule ceased to have this meaning and value, and the general use of the indicator and other means of ascertaining the actual power developed by the engine when working have caused the decline in the use of the expression. But there remains the need for some means of expressing in simple figures the size of a reciprocating engine for purposes of comparison and commerce. There are also other and technical demands for such a denomination, as will be shown.

Thus, for many years, a marine engine was expected to indicate about five times its nominal horse-power. The rule then in use was as follows:—

$$\text{N.H.P.} = \frac{\text{Sum of squares of piston diameters}}{\text{30 to 33 (according to district)}},$$

boiler pressure, piston speed, &c., being left entirely out of consideration. It is, of course, hardly necessary to say that such a rule was quite useless for any scientific purpose; whilst, even for commercial purposes, it gave only a very imperfect idea of the relative values of different engines. Its use, however, still survives in some districts, the divisor being 30, the normal stroke ·618 of diameter of L.P. cylinder, and the normal heating surface 16 square feet per N.H.P.

The matter was still in this chaotic condition when, in 1883, Mr Seaton, in his "Manual of Marine Engineering," suggested the use of E.H.P., or **Estimated Horse-power**, which is now calculated in the following way: —

Rule 1. *Seaton's.* $\text{E.H.P.} = \dfrac{D^2 \times \sqrt{P} \times S \times R}{Z}$

D is the diameter of L.P. cylinder and S the stroke of piston, both in inches; P is the absolute boiler pressure; R, the revolutions per minute.

For naval ships with overloads,	Z = 85,000.
,, short passage express steamers,	Z = 91,000.
,, long ,, ,,	Z = 94,500.
,, passenger cargo steamers,	Z = 97,000.
,, cargo steamers,	Z = 105,000.

Rule 1a. *Seaton's.* $\text{E.H.P.} = \dfrac{D^2 \times WP \times S \times R}{(r+2) \times 140{,}000}$

WP is the working or boiler pressure.
r is the ratio of the L.P. to the H.P. cylinder capacity.

These formulæ give a very close approximation to the horse-power actually indicated when working at full speed.

In 1888, the North-East Coast Institution of Engineers and Shipbuilders proposed the following very complete formulæ:—

Rule 2. *N.E.C. Institution.* $\text{N.I.H.P.} = \dfrac{(D^2\sqrt[3]{S} + 3H\sqrt[3]{P})}{100}$

Where N.I.H.P. = Maximum normal indicated horse-power, on loaded trial trip, of surface-condensing screw engines, working at any pressure between 50 and 250 lbs., under "normal" conditions.

D = Diameter of L.P. cylinder, in inches (if more than one, D^2 must equal sum of squares).

S = Stroke, in inches.

P = Working pressure, in lbs., above atmosphere.

H = Heating surface of boilers in sq. feet.

P_m = Mean pressure, in lbs., referred to L.P. cylinder.

The conditions assumed as "normal" are as follows:—That

(1) Steam of all pressures is expanded down to the same terminal pressure;

(2) Expansion is effected with same degree of efficiency for all pressures;

(3) Piston speeds are proportional to cube roots of strokes, and, further, actual loaded trial-trip piston speed may be taken as $144\sqrt[3]{S}$;

(4) In all cases where relative proportions of engine and boiler prevent (1) being fulfilled without violating (3), the coal consumption will not be affected, but will be constant for the same boiler pressure;

(5) Boilers are of usual proportions and construction, and the horse-power proportional to heating surface (H), and to cube root of pressure ($\sqrt[3]{P}$); and further, actual loaded trial-trip horse-power may be taken as $\frac{H\sqrt[3]{P}}{16}$;

(6) Efficiency of engine mechanism is constant, and the propeller such that engines will utilize boiler power, referred to in (5), in the manner prescribed in (3) and (4).

As a result of (1) and (2), it follows that mean pressure referred to L.P. cylinder (P_m) may be assumed as proportional to cube root of boiler pressure ($\sqrt[3]{P}$), and further, that its actual loaded trial-trip value may be taken, without sensible error, as $5{\cdot}6\sqrt[3]{P}$.

The normal relation between engines and boilers is expressed by the equation.

$$H = \frac{D^2\sqrt[3]{S}}{3{\cdot}25}.$$

The results obtained by the Rule 2 for N.I.H.P., if divided by 6, give quantities very near those found by the old nominal horse-power rule.

It is also claimed that,—for machinery of the same type and design, proportions and arrangement, built of similar materials, under similar circumstances, to similar factors of safety, and not differing very widely in size,—the weights and costs will vary almost exactly as N.I.H.P.

For paddle engines the same rule may be used, with the co-efficients

altered to suit the piston speeds usual for these engines. Assuming that under (4), piston speed of paddle engines may be taken at $90\sqrt[3]{S}$, the rule may, without sensible error, be written,—

Rule 3. *Paddle Engine.* $$\text{N.I.H.P.} = \frac{(D^2\sqrt[3]{S} + 5H)\sqrt[3]{P}}{160}$$

and the normal relation between engines and boilers will be expressed by,—

$$H = \frac{D^2\sqrt[3]{S}}{5{\cdot}2}.$$

The "Standard practice" formerly obtaining on the North-East Coast of England for triple compound engines working with steam of 160 lbs. pressure was as in Table I. To-day N.H.P. is not used there; in its place an estimated I.H.P. is taken for cargo steamer engines designed on Standard Rules (*vide* Appendix L) for a working pressure of 180 lbs. This I.H.P. is $= D^2 \times S \times N \div 700$.

Where D is the diameter of the L.P. cylinder in inches, S is the stroke in feet, and N the revolutions per minute, which by rule $= 32(S + 4) \div S$.

Taking the stroke in inches as S, and combining these, then estimated I.H.P. $= D^2(S_1 + 4) \div 262{\cdot}5$.

Table I.—Sizes of Cylinders, &c., and corresponding N.H.P.

N.H.P.	Diars. of cyls. in inches.			Stroke in ins.	N.H.P.	Diars. of cyls. in inches.			Stroke in ins.
	H.P.	M.P.	L.P.			H.P.	M.P.	L.P.	
20	8	13	22	18	140	19	31½	51½	36
30	9½	15½	26	21	150	20	32½	53	36
40	11	18	30	21	160	20	33	54	39
50	12½	20	33	24	170	20½	34	56	39
60	13½	22	36	24	180	21	35	57	39
70	14	23	38	27	190	21½	36	59	39
80	15½	25	40	27	200	22	36½	60	42
90	16	26	43	30	225	23	38	63	42
100	16½	27	45	30	250	24½	40	66	45
110	17	28	46	33	275	25	41½	69	45
120	18	29	48	33	300	26	43	71	48
130	18½	30	50	33	...	...	...	...	...

Rule 5. *Board of Trade Rule* for registration purposes:—

$$\text{N.H.P.} = \frac{(3H + D^2\sqrt[3]{S})\sqrt[3]{P}}{700}$$

where H = heating surface of main boilers in square feet.

D^2 = square of diameter of low-pressure cylinder, or sum of squares of diameters of cylinders in non-compound engines, measured in inches.

S = length of stroke of engines in inches.

P = pressure of main boilers.

Rule 6. *Lloyd's Rule* for determining amount of survey fees, &c., is as follows:—

$$\text{N.H.P.} = \frac{P+C}{K}\left(\frac{D^2\sqrt{S}}{100}+\frac{H}{15}\right)$$

where D is diameter of L.P. cylinder in inches; S, stroke in inches; H, heating surface in sq. feet; P, working pressure in lbs. per sq. inch; and

$$C = \begin{cases} 340 & \text{where boiler pressure is below 160 lbs.} \\ 590 & \text{,, ,, 160 lbs. or above.} \end{cases}$$

$$K = \begin{cases} 1000 & \text{,, ,, below 160 lbs.} \\ 1500 & \text{,, ,, 160 lbs. or above.} \end{cases}$$

If the boilers are fitted for other than natural draught, $\frac{H}{12}$ is to be substituted for $\frac{H}{15}$. H includes surfaces of tubes, of back tube-plates, and of furnaces and combustion chambers down to level of fire bars.

Since the capacity of the L.P. cylinder is the true measure of the size of the engine, and as the other cylinders have a relation to the L.P. which does not vary largely in any but naval engines; and, further, as the standard stroke in general practice varies with the diameter of cylinders, the following gives a good criterion of the size of the engine, which may be usefully employed for various purposes:—

Rule 7. *Seaton's* (2). $\text{N.H.P.} = D \times S \div X$,

where D and S are as before the diameter of L.P. cylinder and stroke in inches, and X a factor which for compound engines is 15, for triples 12·5, and for quadruples 10·5, and extension of this is—

Rule 8. *Seaton's* (3). $\text{N.H.P.} = \frac{D \times S \times \sqrt{P}}{150}$ for all engines

where P is the absolute boiler pressure.

Table II.—Nominal Horse-power of Triple Compound Engines ($D \times S \div 12{\cdot}6$).

Diameter of Low-Press. Cylr.	Strokes in inches.																	
	21	24	27	30	33	36	39	42	45	48	51	54	57	60	63	66	69	72
ins.																		
30	50	56	64	70	77	84	91	98	..	..	..	..	..	..	..	..	..	..
32	53	61	68	76	84	92	100	107	..	..	..	..	..	..	..	..	..	..
34	56	64	72	81	89	97	105	113	121	..	..	..	..	..	..	..	..	..
36	60	68	77	85	94	102	111	120	128	137	..	..	..	..	..	..	..	..
38	63	71	80	89	98	107	116	126	135	144	153	..	..	..	..	..	..	..
40	66	76	85	95	104	114	123	133	142	152	161	171	..	..	..	..	..	..
42	..	80	90	100	110	120	130	140	150	160	170	180	190	..	..	..	..	..
44	..	83	94	104	115	125	136	146	157	167	178	188	198	209	..	..	..	..
46	..	87	98	109	120	131	142	153	164	175	186	197	208	219	..	..	..	..
48	..	..	102	114	125	136	148	159	170	182	193	205	216	228	..	..	..	..
50	..	..	107	119	130	142	154	166	178	190	202	214	226	238	..	..	..	..
52	..	..	..	123	136	148	161	173	185	198	210	222	235	247	..	..	..	..
54	..	..	..	128	141	154	167	180	192	205	218	231	244	257	..	..	..	..
56	..	..	..	133	146	160	173	186	200	213	226	240	253	266	..	..	..	..
58	..	..	..	..	151	165	179	193	207	221	234	248	262	276	..	..	..	..
60	..	..	..	..	157	171	185	200	214	228	242	257	271	285	..	..	..	..
62	..	..	..	..	..	177	191	206	221	236	250	265	280	295	..	..	..	..
64	..	..	..	..	..	182	198	213	228	243	259	274	289	304	320	..	..	..
66	..	..	..	..	..	188	204	220	235	251	267	282	298	314	330	..	..	..
68	..	..	..	..	..	..	210	226	242	258	274	290	306	323	339	..	..	..
70	..	..	..	..	..	..	216	233	250	266	283	300	316	333	350	..	..	..
72	..	..	..	..	..	..	222	240	257	274	291	308	325	342	360	377	..	..
74	..	..	..	..	..	..	..	246	264	282	299	317	334	352	370	387	..	..
76	..	..	..	..	..	..	..	253	271	289	307	325	343	362	381	399	..	..
78	..	..	..	..	..	..	..	260	278	297	315	334	352	371	390	408	..	..
80	..	..	..	..	..	..	..	266	285	304	323	342	361	380	400	419	439	..
82	..	..	..	..	..	..	..	..	292	312	331	351	371	390	410	429	449	..
84	..	..	..	..	..	..	..	..	300	320	340	360	380	400	420	440	460	..
86	..	..	..	..	..	..	..	..	307	327	348	368	389	409	430	450	470	491
88	..	..	..	..	..	..	..	..	314	335	356	377	398	419	440	461	481	502
90	..	..	..	..	..	..	..	..	..	342	364	385	407	428	450	471	492	514
95	..	..	..	..	..	..	..	..	.	362	384	407	429	452	475	497	520	543
100	..	..	..	..	..	..	..	..	..	381	404	428	452	476	500	523	547	571

Table III.—Nominal Horse-power of Quadruple Compound Engines.

Diameter of L.P. Cylinders.	Strokes.													
	24 ins.	27 ins.	30 ins.	33 ins.	36 ins.	39 ins.	42 ins.	45 ins.	48 ins.	51 ins.	54 ins.	57 ins.	60 ins.	63 ins.
ins.														
36	82	92	103	113	123	133	144	154	..	..	..	..	..	..
38	86	97	108	119	130	141	152	163	..	..	..	..	..	..
40	91	102	114	125	137	148	160	171	..	..	..	..	..	..
42	96	108	120	132	144	156	168	180	192	..	..	..	..	..
44	..	113	125	138	150	163	176	188	201	..	..	..	..	..
46	..	118	131	144	157	170	184	197	210	..	..	..	..	..
48	..	123	137	150	164	178	192	205	219	233	..	..	..	..
50	..	..	142	157	171	186	200	214	228	242	257	..	..	..
52	..	..	148	163	178	193	208	223	237	252	267	..	..	..
54	..	..	154	169	185	200	216	231	247	262	277	293	308	..
56	..	..	160	176	192	208	224	240	256	272	288	304	320	..
58	..	..	..	182	198	215	232	248	265	281	298	315	331	..
60	..	..	..	188	205	223	240	257	274	291	308	326	343	..
62	..	..	..	194	212	230	248	265	283	301	318	337	354	..
64	..	..	..	201	219	238	256	274	293	311	329	348	366	..
66	..	..	..	207	226	245	264	283	302	321	339	358	377	396
68	..	..	..	213	232	252	272	291	311	330	349	369	388	408
70	..	..	..	220	240	260	280	300	320	340	360	380	400	420
72	..	..	..	..	247	267	288	308	329	349	370	391	411	432
74	..	..	..	..	254	275	296	317	338	359	380	402	423	444
76	..	..	..	..	260	282	304	325	347	369	390	412	434	456
78	..	..	..	..	267	290	312	334	356	379	401	423	446	468
80	..	..	..	..	274	297	320	342	365	388	411	434	457	480
82	..	..	..	..	281	304	328	351	374	398	421	445	468	492
84	..	..	..	..	287	311	336	359	383	407	431	456	479	504

Summary Rules for Estimating Horse-power.

1. Estimated I.H.P. $= \frac{D^2\sqrt{P+15} \times R}{Z} \times \frac{S}{12}$ (Seaton, *v.* "Manual," p. 197).

2. Nominal I.H.P. (Screw Engines) $= \frac{(D^2\sqrt{S}+3H)\ \sqrt[3]{P}}{100}$ (N.E. Coast Inst. E. and S.).

3. Nominal I.H.P. (Paddle Engine) $= \frac{(D^2\sqrt{S+5H})\sqrt[3]{P}}{160}$ (N.E. Coast Inst. E. and S.).

4. Estimated I.H.P. $= D^2(S_1+4) \div 262{\cdot}5$ (N.E. Coast Engineers, 1917).

5. Nominal H.P. $(3H + D^2 \sqrt[3]{S}) \times \sqrt[3]{P} \div 700$ (Board of Trade).

6. Nominal H.P. $= \frac{P+C}{K}\left(\frac{D^2\sqrt{S}}{100} + \frac{H}{15}\right)$ (Lloyd's).

7. Nominal H.P. $= D \times S \div X$ (Seaton, *v.* "Manual," p. 196).

8. Nominal H.P. $= D \times S \times \sqrt{P+15} \div 150$ (Seaton).

d is the diameter of the H.P. cylinder, d_1 that of the first intermediate, etc., and D that of the low pressure; S the stroke of piston; all in inches. P the load pressure on safety valves in lbs. per sq. inch; H the heating surface of the boilers in sq. feet; C and K are 340 and 1000 respectively when the boiler pressure is under 160 lbs., and 590 and 1500 when above 160. R the revolutions per minute; and X is 15 for compound engines, 12·5 for triple, and 10·5 for quadruple. Z varies from 85·000 to 105·000 (*v. ante*).

Table IIIa.—Horse-power in ft.-lbs. of Foreign Countries.

Country.	British ft.-lbs. per minute.	Ratio of foreign to British.	Kilogrammetres per second.
France	32,552	0·9865	75·000
Prussia . . .	32,689	0·9900	75·325
Austria . . .	33,034	1·0010	76·119
Saxony . . .	32,568	0·9869	75·045
Hanover . . .	32,705	0·9905	75·361
German Empire . .	33,000	1·000	76·041

Indicated Horse-power.—The indicated horse-power of an engine may be defined as the measure of work done in the steam cylinder, as deduced from the indicator diagrams, and is equal to (area of piston in square inches × mean pressure in lbs. per square inch × number of feet travelled through by piston, per minute) ÷ 33,000; or,—

Rule 9. $$\text{I.H.P.} = \frac{A \times P \times S}{33{,}000}$$

Piston speed, S, = stroke in feet × 2 × number of revolutions per min.

In the case of engines having more than one steam cylinder, the I.H.P. of each cylinder is determined separately, and the sum of these is the I.H.P. of the engine.

Where accuracy is required, the sectional area of piston rod should be deducted from the areas of piston.

Shaft Horse-power is that transmitted by the shaft and is measured by the torque on it. If T_q is the torque in lbs., θ the angle of torsion on a length of shaft *l* inches, whose diameter is *d* inches, then

Rule 10. $$\text{S.H.P.} = \frac{\theta^0 \times d^4 \times \text{revs.}}{3{\cdot}27 \times l}.$$

Nett Horse-power is that required to overcome the resistance of the ship only, then if R is the resistance in lbs., S the speed in knots per hour, we get

Rule 11. $$\text{Nett H.P.} = \frac{\text{R} \times \text{S} \times 6080}{33{,}000 \times 60} = \frac{\text{R} \times \text{S}}{325}.$$

Tow-rope Horse-power is that required to tow the ship without a propeller, T is the tension on tow-rope in lbs.

Rule 12. $$\text{T}_r\text{H.P.} = \frac{\text{T} \times \text{S}}{325}.$$

Propeller Horse-power is that received by the propeller.

Thrust Horse-power is that delivered by the propeller for the propulsion of the ship, so that T_s is the thrust in lbs.

Rule 13. $$\text{T}_s\text{H.P.} = \frac{\text{T}_s \times \text{S}}{325}.$$

Summarising.

(1) I.H.P. is the gross power generated.

(2) S.H.P. is the nett power delivered by the engine to the shafting.

(3) P_rH.P. is that received by the propeller, being S.H.P. *minus* the loss at the thrust and other bearings and in the stern tubes.

(4) T_sH.P. is that delivered to drive the ship, being P_rH.P. less that lost by friction, eddies, etc., in its working.

(5) T_rH.P. is the thrust H.P. less that lost by the augmented resistance set up by the screw, and is practically nett H.P.

To determine the mean pressure from an indicator diagram the following is general practice:—

Let fig. 1 represent a pair of diagrams from the L.P. cylinder of a compound engine. Draw two perpendiculars to the atmospheric line AB, one at each end of and touching the diagrams; divide the space between them into ten equal parts, placing the division marks so that there shall be half a space at each end, and draw a vertical ordinate through each mark; measure off the breadths of the diagrams at each ordinate to the scale corresponding to the indicator spring and figure them on ends of ordinates as shown, keeping the figures referring to each diagram in a separate column.

The sum of each divided by ten gives two mean pressures,—one of which refers to each side of the piston,—and the mean of these two is the mean pressure required.

Planimeter.—Where there are many diagrams to be calculated, it is quicker to use a planimeter in place of the method given above. The method is as follows:—Measure the area of the figure by means of the instrument, and divide it by the length AB, when the quotient will be the mean breadth in inches; and this, multiplied by the

"scale" of the spring used (the number of pounds required to compress it one inch) will give the mean pressure required.

The Coffin averaging instrument is a form of planimeter specially designed for dealing with indicator diagrams. It leaves on the card two needle pricks, the distance between which is the mean breadth of

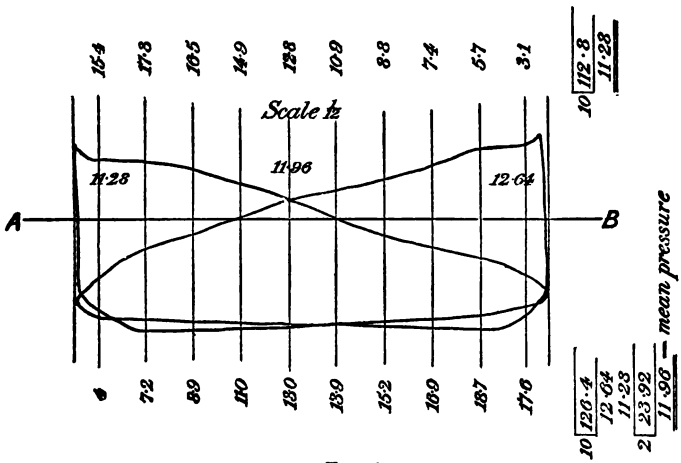


FIG. 1.

the figure,—and thus performs mechanically the process of dividing area of figure by length.

The following equivalents may be useful in calculations connected with the above:—

Pounds per sq. inch × ·07 = Kilogrammes per sq. centimetre.
Kilogrammes per sq. centimetre × 14·22 = Pounds per sq. inch.

Foot-pounds × 7·233 = Kilogrammetres.
Kilogrammetres × ·138 = Foot-pounds.

Horse-power × 1·0139 = Chevaux.
Chevaux × ·9863 = Horse-power.

See also "Tables of Pounds per square inch and Kilogrammes per square centimetre," Table CLXXXVIII.

The Continental "Cheval" is equal to 4500 kilogrammetres, or 32,549 foot-pounds per minute, as against 33,000 foot-pounds per minute,—the value of the English horse-power."

The following Table will considerably facilitate the computation of indicated horse-power:—

Table IV.—Constant Multipliers for I.H.P.

Constant = $\frac{\text{area of cylinder}}{33{,}000}$; and I.H.P = constant × mean press. × piston speed.

Diameter of Cylinder.	Constant.	Diameter of Cylinder.	Constant.	Diameter of Cylinder.	Constant.	Diameter of Cylinder.	Constant.
6	·00085	16½	·00648	34	·02751	60	·08569
¼	·00092	¾	·00668	½	·02833	61	·08856
½	·00100	17	·00688	35	·02916	62	·09149
¾	·00108	¼	·00708	½	·02999	63	·09447
7	·00116	½	·00728	36	·03084	64	·09748
¼	·00125	¾	·00749	½	·03171	65	·10055
½	·00134	18	·00771	37	·03258	66	·10368
¾	·00143	¼	·00792	½	·03347	67	·10684
8	·00152	½	·00814	38	·03437	68	·11005
¼	·00162	¾	·00836	½	·03528	69	·11332
½	·00172	19	·00859	39	·03620	70	·11663
¾	·00182	¼	·00882	½	·03713	71	·11998
9	·00192	½	·00905	40	·03808	72	·12339
¼	·00203	¾	·00928	½	·03904	73	·12683
½	·00214	20	·00952	41	·04001	74	·13033
¾	·00226	½	·01000	½	·04099	75	·13388
10	·00238	21	·01049	42	·04198	76	·13748
¼	·00250	½	·01100	½	·04299	77	·14112
½	·00262	22	·01152	43	·04401	78	·14481
¾	·00275	½	·01205	½	·04504	79	·14854
11	·00288	23	·01259	44	·04608	80	·15232
¼	·00301	½	·01314	½	·04713	81	·15615
½	·00314	24	·01371	45	·04820	82	·16003
¾	·00328	½	·01428	½	·04927	83	·16398
12	·00342	25	·01487	46	·05036	84	·16795
¼	·00357	½	·01547	½	·05146	85	·17198
½	·00372	26	·01609	47	·05257	86	·17604
¾	·00387	½	·01671	½	·05370	87	·18016
13	·00402	27	·01735	48	·05483	88	·18432
¼	·00417	½	·01800	½	·05598	89	·18853
½	·00433	28	·01866	49	·05714	90	·19280
¾	·00449	½	·01933	½	·05832	91	·19710
14	·00466	29	·02001	50	·05950	92	·20146
¼	·00483	½	·02071	51	·06191	93	·20587
½	·00500	30	·02142	52	·06436	94	·21030
¾	·00517	½	·02214	53	·06685	95	·21480
15	·00535	31	·02287	54	·06940	96	·21937
¼	·00553	½	·02362	55	·07200	97	·22394
½	·00571	32	·02437	56	·07464	98	·22859
¾	·00590	½	·02513	57	·07733	99	·23328
16	·00609	33	·02592	58	·08006	100	·23799
¼	·00628	¼	·02671	59	·08285	101	·24280

When the effective pressure on the piston at each point in the stroke is required,—as for instance, to calculate the twisting moment on the crankshaft, — diagrams should be constructed from the indicator diagrams, as follows :—

First, draw the line of no pressure, CD. Then, dealing with one stroke at a time, the curve EFG represents the varying pressures on one side of the piston, whilst the opposing pressures are represented by the curve JKB, which forms a part of the diagram from the other side of the piston.

Draw any ordinate FH, and set off HL = HF - HK ; then L is a point in the required resultant diagram in which any number of other points

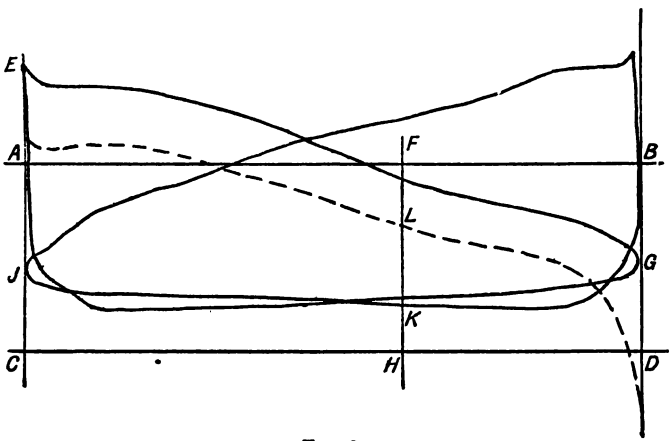


FIG. 2.

may be found in a similar way ; at the point H, in the stroke CD, the effective pressure is HL. When the quantity corresponding to HF - HK is minus, it must be set off below the line of no pressure.

The pipes leading from the ends of the cylinder to the indicator should be large, short, of equal length, and as free from bends as possible ; otherwise, there will be loss of area in the diagram, and the apparent I.H.P. less than that really developed.

These pipes vary, in common practice, from $\frac{3}{4}$-inch to 1-inch diameter, according to their length and the piston speed, but are still, beyond doubt, the cause of very perceptible loss of area.

To make accurate tests of engines, to determine water consumption per I.H.P., &c., a separate indicator should be fitted direct to each end of each cylinder. This practice should always be adopted where anything like accuracy is required.

N. B.—The indicator shows only *differences* between the pressures of the steam and of the atmosphere, and not absolute pressures.

To ascertain the weight of steam accounted for by any diagram, take a point A in the expansion curve of the diagram (fig. 3), just before release, and measure the absolute pressure there; then take another point B in the compression curve, and measure the absolute pressure here also; from Table CLXXIX. ascertain the weight of a cubic foot of steam at each of the pressures AZ and BZ.

Now calculate the volume, in cubic feet, swept by the piston while

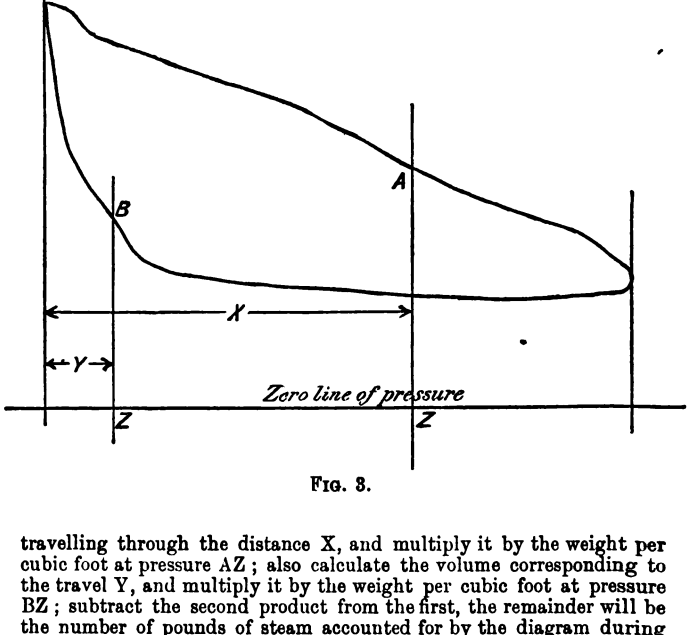


FIG. 3.

travelling through the distance X, and multiply it by the weight per cubic foot at pressure AZ; also calculate the volume corresponding to the travel Y, and multiply it by the weight per cubic foot at pressure BZ; subtract the second product from the first, the remainder will be the number of pounds of steam accounted for by the diagram during the stroke.

A similar calculation from the other diagram of the pair will give the amount of steam accounted for during the return stroke, and the sum of the two—multiplied by the number of revolutions—the amount per minute or per hour.

The clearances need not be considered in these calculations if the

points A and B be taken at the same distance above the zero line of pressure.

This method is mainly useful in determining condensation and re-evaporation that occur during passage of steam through a series of cylinders. No conclusion as to economy can be derived from diagrams only. They tell nothing about water that may be present in cylinders. Feed water must be measured to determine real economy of engine.

Shaft Horse-power is that transmitted by a shaft, and can be calculated from the *torque* as follows :—

T is the twisting moment or *torque* in inch-pounds.

R, the revolutions made per minute by the shaft.

The work performed per revolution $= \frac{2\pi \times T}{12}$ or $0{\cdot}5236$ T.

Rule 14. $\text{S.H.P.} = \frac{0{\cdot}5236\ T \times R}{33{,}000} = \frac{T \times R}{63{,}000}$.

The torque on any shaft can be calculated from the angle of twist or torsion by means of the following formulæ :—

a is the arc at a radius r of the angle of torsion θ, $\frac{a}{r} = \beta$; l is the length and d the diameter of shaft under observation.

$$\beta = \frac{10{\cdot}2 \times T \times l}{M_r \times d^4} \quad \text{(Rankine)}.$$

$$\frac{\theta}{360} = \frac{a}{2\pi r} \text{ or } = \frac{\beta}{2\pi}.$$

That is $$\beta = \frac{\theta \times 2\pi}{360} = \frac{\theta}{57{\cdot}3}.$$

Then $$\frac{\theta}{57{\cdot}3} = \frac{10{\cdot}2 \times T \times l}{M_r \times d^4},$$

or (i) $$\theta = \frac{584 \times T \times l}{M_r \times d^4} \text{ for solid shafts,}$$

(ii) $$\theta = \frac{584\ T \times l}{M_r(d^4 - d_1^4)} \text{ for hollow shafts when } d_1 \text{ is the diameter of the bore.}$$

That is $$T = \frac{\theta \times d^4 \times M_r}{584 \times l}.$$

M_r is the modulus of stiffness or rigidity of the material, which for steel generally is 10 to 12 millions. With steel shafts of best make, experiments have shown the value of M_r to be 11,750,000 for solid, and

12,150,000 for hollow. In everyday practice, 11,250,000 is taken for solid steel shafts:—

then $$T = \frac{\theta \times d^4 \times 11{,}250{,}000}{584\,l} = 19{,}264\frac{\theta \times d^4}{l}.$$

Substituting this value of T in the formula for S.H.P.

Rule 15. $$S.H.P. = 19{,}264\,\frac{\theta \times d^4}{l} \times \frac{R}{63{,}000} = \frac{\theta \times d^4 \times R}{3{\cdot}27 \times l}$$ solid shafts.

Rule 15a. $$S.H.P. = \frac{\theta(d^4 - d_1{}^4)R}{3{\cdot}27 \times l}$$ for hollow shafts.

Torsion Meters are the instruments employed to indicate the angular displacement of a definite length of shaft (usually 40 inches), when the shaft is receiving and transmitting power. They are all designed on the same fundamental principle, and consist of two cast-iron sleeves made in halves so as to go on the shaft, and be keyed to it a definite distance apart; one is short and the other long; each has a disc of considerable diameter at the ends adjoining, so that as the shaft twists the discs' edges are displaced with respect to one another by the amount a, which may be used instead of $\theta°$, if desired, for calculating S.H.P. thus:—

Rule 15b. $$S.H.P. = \frac{17{\cdot}5 \times d^4 \times R \times a}{r \times l}.$$

There are many forms of torsion meter which may be classed by the methods adopted for indicating θ or β:—

(i) Those having differential levers whereby the distortion is magnified, such as Professor Föttinger's.

(ii) Those which accomplish the same end by wheel and pinion gearing, as in the Denny-Edgecombe machine.

(iii) Those without sleeves, but with counter shafts coupled at their extreme ends to the shaft by driving gear and to one another by screw and nut, which cause longitudinal displacement, which is magnified by a lever; as in the Collie meter.

(iv) Those which magnify the arc of the angle of torque by means of a beam of light reflected from a mirror, which is twisted through angle θ; as in the Hopkinson-Thring instrument, which entails the use of the least amount of gearing.

(v) Others, as the Bevis-Gibson, where a beam of light is made to pass through slits in discs at a considerable distance apart; which becomes blinded on angular distortion taking place, and visible again after an angular movement of the sighting hole corresponding to θ.

The registering is made mechanically and automatically by the *Föttinger* on a sheet of paper secured on a fixed drum surrounding the

shaft, so that the magnitude of the angle or arc can be measured at any point by the distance from the base line made when the shaft is transmitting no power.

The *Denny-Edgecombe* machine shows the distortion on a dial which, however, is in motion on the shaft, but by a most ingenious electrical arrangement its indications are given on another dial fixed in any convenient part of the ship. It can be and often is made to automatically register its indications on a diagram, which show us the vagaries in torque of a reciprocator, or the mean torque of it or of a turbine.

The *Hopkinson-Thring* mirrors throw a point or line of light on a fixed scale, from which at any time the angle may be read off; it also registers its own zero line continuously, and by an arrangement can indicate the variations in torque of a reciprocator.

The *Bevis-Gibson* measures the torque by the amount of displacement required in the registering apparatus to bring the light holes in line.

A simple formula for each ship can be made and used for S.H.P. by simply multiplying it by θ and revolutions; in fact, a table may be made by including the revolutions, so that from it a co-efficient may be taken and multiplied by θ only. Thus if l is 40 and the diameter of the shaft 12 ins., then

Rule 16. S.H.P. $= 158{\cdot}5\ \theta \times R$.

And if at 400 revolutions θ is $0{\cdot}165°$, S.H.P. $= 10{,}461$.

EFFICIENCY OF MARINE MACHINERY.

Efficiency is expressed by the fraction of the whole work or energy generated or supplied that is usefully employed for the purpose for which it was designed. Hence,

Efficiency of the boiler is the available heat transmitted to the engine as a fraction of that given out by the fuel.

Efficiency of the steam is the fraction of the total heat of evaporation that is capable of being converted into work.

The thermal efficiency of an engine is expressed by the fraction of the work capable of being done by the steam or gas that actually is done by it.

The mechanical efficiency is the fraction of the work developed in the generator that is actually passed on from it for external consumption.

The general efficiency of a steam or gas engine is the fraction that the output of energy is of that available from the fuel.

Propeller efficiency is the fraction of the power delivered to it that is devoted to thrust power.

Propulsive efficiency is the relation of the nett or tow-rope H.P. to the gross H.P. developed in the propelling engines; strictly speaking, it should be taken to the H.P. delivered to the propeller shafting, which can now be ascertained by torsion meters.

Boiler efficiency is measured by taking the number of pounds of ste actually evaporated as the numerator and the thermal value of the f divided by the total heat of evaporation under the same conditions denominator. For example, the average thermal value of the ordin good Welsh steam coal is 15,500 B.T.U. The total heat of evaporati from 110° and at 341° (120 lbs. pressure abs.) is 1108 B.T.U., then

$$\text{greatest possible evaporation} = \frac{15,500}{1108} = 14 \text{ lbs.}$$

A boiler using this fuel evaporates 10·8 lbs. of steam per lb. of fuel

$$\text{The efficiency} = \frac{10{\cdot}3}{14} \text{ or } 0{\cdot}736.$$

Efficiency of steam by Rankine's formula is as follows:—

$$E = \frac{\text{effective mean pressure in cylinders}}{\text{mean pressure} + \text{rate of expansion} \times pr^{-1}/_{16}}.$$

Thermal efficiency is expressed, then, where T_1 is the absolute tempers ture at entry and T_2 that at exit or exhaust.

Rule 17. Then thermal efficiency $= \frac{T_1 - T_2}{T_1}$.

For example, an engine works with steam at 180 lbs. pressure abs. at the H.P. valve and rejects into a condenser with a vacuum of 28 ins. Here $T_1 = 373° + 461° = 834°$.

$T_2 = 90° + 461° = 551°$.

$$\text{Thermal efficiency} = \frac{834 - 551}{834} = 0{\cdot}315.$$

The Maximum Thermal Efficiency of any Condensing Steam Engine with a Temperature of 60° F. in the Condenser.

Working press. abs.	40	80	100	120	140	160	180	200	220	240	260	280
Efficiency . .	0·284	0·326	0·339	0 350	0·360	0·368	0·375	0·381	0·387	0·393	0·398	0·404

The mechanical efficiency of engines generally is ascertained by comparing the output H.P. as measured by a brake or other mechanical means. This is not possible with engines of great power. Since the introduction of the turbine as a marine motor, the torsion meter has been invented and used to indicate S.H.P. in reciprocators. Before this experiments had been made to ascertain the friction H.P. by running engines without their propellers at various rates of revolution and deducting it from the I.H.P. at the corresponding revolutions coupled up. The following has been ascertained by one or other of these methods:—

Rule 18. Mechanical efficiency $= \frac{\text{S.H.P. or B.H.P.}}{\text{I.H.P.}}$.

Large quadruple engines by Denny Bros., efficiency $\frac{\text{S.H.P.}}{\text{I.H.P.}} = 92$ to 94 per cent.

Triple compound engines 900 I.H.P. by Central Marine Eng. Co., $\frac{\text{I.H.P.} - \text{F.H.P.}}{\text{I.H.P.}} = 95$ per cent.

Several large engines (triples) made in Germany showed an efficiency from 88·5 per cent. to 93·5 in the largest (4500 I.H.P.).

Triple compound engines, 300 I.H.P., by Earles S. & E. Co. showed 84·9 per cent., and similar engines 480 I.H.P. 90·2.

Compound engines of a torpedo boat by Mr A. F. Yarrow, 255 I.H.P., showed by dynamometric trials 92·3 per cent., and as much as 76·6 per cent. when developing only 38·5 I.H.P.

Triple compound engines with *forced lubrication* by Belliss & Morcom gave by electric output and brake trials 93·15 per cent. at 429 I.H.P., and as much as 98·8 per cent. at 218 I H.P.

Marine engine losses vary very nearly with the revolutions but the power as the cube, the efficiency at low speeds is then less than at high ones with the same engine; further, the efficiency of small engines, running on similar conditions as to pressure, rates of expansion, etc., as large ones, is lower. On the other hand, the smaller the engine is for the power to be developed the higher will be its efficiency mechanically.

General efficiency of steam engines is gauged by the water or steam consumption per I.H.P. The following schedule gives the greatest possible output of work by a pound of steam under various conditions, and the corresponding horse-power:—

If a steam engine consumes x lbs. of steam per H.P. hour, or $\frac{x}{60}$ per H.P. minute, then $x = 60$ I.H.P., or, 1 lb. produces $\frac{60}{x}$ I.H.P.

If X is the greatest possible output under similar conditions.

Rule 19. General efficiency of engines $= \frac{60}{x} \div X$ or $\frac{60}{Xx}$.

General efficiency of all engines must, however, be commercially dealt with differently, as fuel is the serious factor in any installation and must form the basis of comparison of a steam with an internal combustion engine. In this case the thermal value of a H.P. is 33,000 ÷ 778 or 42·4 B.T.U. The value per hour is 2544 B.T.U. The Welsh coal, with its calorific value of 15,500 B.T.U., is equal to $\frac{15{,}500}{2544}$ or 6·10 H.P.

If an engine uses 1·3 lbs. per I.H.P. hour,

its efficiency will be $\frac{1}{6{\cdot}1 \times 1{\cdot}3} = 0{\cdot}126$ or only 12·6 per cent.

The oil engine using 0·45 lb. of fuel, whose calorific value is 18,8 B.T.U., will have an efficiency as ascertained in the same way—

$$1 \div \frac{18{,}000}{2544} \times 0{\cdot}45 = 0{\cdot}30 \text{ or } 30 \text{ per cent.}$$

Rule 20. General efficiency of machinery installations
$= 2544 \div$ B.T.U. value of fuel $\times$ weight consumed per H. P. hour.

From observations made at the tests of electric generating engine it is found that at constant revolutions the power required to overcom the friction of the engine itself is the same whatever the load may be and with varying revolutions and varying loads the friction *per revolution* is nearly constant; it is therefore probable that with marin engines total friction varies nearly directly as the revolutions.

Now, as the total I.H.P. varies as the cube of the revolutions, i will be seen that the mechanical efficiency of a particular marine engine is at a maximum at full speed.

Froude's Method.—As there is considerable difficulty in experimentally determining the power absorbed in overcoming the friction of a marine engine, the following graphic method is of interest:—

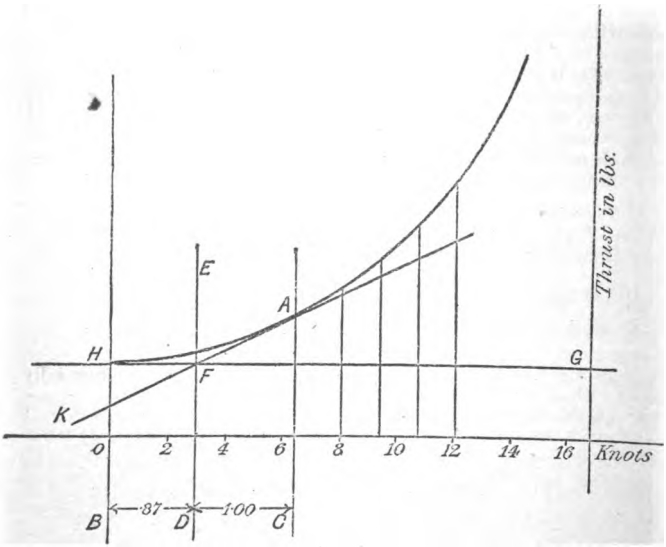


FIG. 4.

A series of progressive trials having been carried out, and the results

carefully recorded—calculate the indicated thrusts (from the formula—Indicated thrust $= \frac{\text{I H.P.} \times 33{,}000}{\text{P} \times \text{R}}$, where P is pitch of propeller in feet, and R revolutions per minute) for each speed, and set them up as ordinates from a base line on which the speeds are set off—as shown in fig. 4. Then, supposing A to be the lowest known point on the curve, draw the tangent KA; divide BC at D so that BC = 1·87 DC, and through D draw the vertical line DE, cutting the tangent in F; and through F draw HG parallel to the base line. The height OH then represents the constant friction of the engine, and the point H is the vertex of the thrust curve which may now be completed.

The lengths of the ordinates intercepted between HG and the thrust curve represent power expended in overcoming ship's nett resistance, augment of resistance due to propeller, and friction of propeller blades, and are proportional to the ship's resistance,

It is also very difficult to determine the frictional resistance of any pair of parts of an engine, owing to the difficulty of reproducing exact conditions of actual work in an experimental apparatus, so here again values can only be estimated from those obtained by experiments made under conditions approximating to those of an actual engine.

The experiments of Tower (made for the Institution of Mechanical Engineers, 1883–91) and of Dewrance (1896) are instructive, as the results obtained were most important; but their tests were all made on journals under constant load, and therefore under very different conditions to those prevailing in the crank bearings of reciprocating engines, and experience has shown the latter to be capable of carrying much greater pressures than bearings subject to constant load.

Tower and Dewrance's experiments may be summarised as follows:—

1. When a bearing is running satisfactorily the surfaces are not in metallic contact, but are separated by a film of oil.

2. Actual measurement by pressure gauges showed that near centre of brass oil was under a pressure somewhat greater than mean pressure per square inch (*i.e.* total load ÷ dia. × length of journal) carried by bearing at the time, and that pressure decreased as ends and sides of bearing surface were approached, and became zero at edges.

3. The drilling of an oil hole at centre of crown of brass allowed oil to be forced out, caused pressure to fall, and diminished load that bearing could carry. These results made it clear that oil should be introduced to bearings of this type where pressure is least, and that no openings should be made near centre of bearing surface, where pressure is greatest. It was found necessary to lubricate experimental journal by means of an oil bath, because, when oil holes in top of brass were used, results were very irregular and frictional resistance was about four times that registered when oil bath was in use.

4. So long as lubrication remained efficient the frictional resistance under varying loads remained constant, *i e.* was unaffected by pressure per square inch to which bearing was loaded.

5. The frictional resistance varied very nearly as the square root of the rubbing velocity.

6. The temperature at which the bearing was run was found to l most important, one experiment, made with lard oil under a pressu of 100 lbs. per square inch, showing that the frictional resistance wa three times greater at 60° than at 120°. Apparently there is a mos suitable temperature for each lubricant at each pressure.

For a temperature of 90° the above results are very accurately expressed by the formula—

$$F = 20C \times \frac{\sqrt{V}}{P},$$

where F = friction factor or co-efficient, V = rubbing velocity in ft. per min., P = nominal pressure in lbs. per sq. in., C = ·0014 for sperm oil, ·0015 for rape, ·0018 for mineral, and ·0019 for olive oil. P had a range of about 100 to 500 lbs. per square inch.

Frictional resistance depends on—(*a*) Velocity of rubbing; (*b*) intensity of pressure; (*c*) temperature; (*d*) lubricant; (*e*) nature of rubbing, *i.e.* continuous or constantly reversed; (*f*) form of surface, *i.e.* curved, like a bearing, or flat; (*g*) material and condition of surface; (*h*) extent of contact, *i.e.* all over a surface, or only on a line; (*j*) method of application of lubricant.

As regards (*g*), Dewrance's experiments appear to show that the composition of the alloys used had, other things being equal, little or no effect on results obtained,—a result difficult to reconcile with common experience to the contrary with ordinary engine bearings.

As regards (*h*), reduction in surface appears to reduce frictional resistance (as in the case of a railway carriage wheel when "skidded"), so that surfaces should not greatly exceed those found to give reasonable length of life under usual conditions.

Dewrance found (*j*) most important, and by careful arrangements succeeded in carrying loads of well over one ton per square inch, whereas with defective (but very usual) arrangements the same bearing would not carry one-tenth of that load. He found it necessary to ease away the brasses at the sides in a very gradual manner and to admit the oil at the sides, or points of least pressure, whence it was carried forward by the revolving journal into the gradually narrowing clearance spaces and towards the points of greatest pressure.

Any sudden contraction of this clearance space, or any ridge tending to scrape the oil from the surface of the journal, was found to lower seriously the load-carrying power of the bearing.

The fact that a crosshead shoe will not usually work satisfactorily with a load of more than about 50 lbs. per square inch, whilst a bearing can easily be made to carry ten times as much, still waits an explanation. Difficulty of lubrication, and the continual reversal of the motion, will no doubt partly explain the anomaly; but, even if all the rubbing surfaces about an engine were of equal smoothness, worked at same temperature, with same lubricant, and at same speed of rubbing, it would still be almost impossible to fix the various values of the friction factor. And as to frictional losses caused by

tightness of piston rings, and of gland packings, and by screwing up of bearings,—they may be anything, and cannot even be estimated.

The efficiency of the same engine may therefore vary greatly at different times, and the frictional resistance of a bearing (say) may be almost anything between that due to the friction of solid on solid and that due to liquid friction. The laws governing the friction of solids are as follows:—

(*a*) The frictional resistance varies directly as the load.

(*b*) It is independent of the extent of surface in contact.

(*c*) It tends to diminish with an increase of velocity above a certain limit.

THE THERMAL AND POWER VALUE OF A POUND OF STEAM.

Maximum Output from 1 lb. of Steam in B.T.U. and Horse-power.

The following Tables—V., VI., and VII.—give the amount of work that is theoretically possible under the different conditions stated. Table VII. is a general one giving the output when steam is expanded from an actual pressure p_1 to a pressure p_2, at which it discharges; this is the case of a turbine during its various stages and throughout. Table V. gives the output when steam is expanded from a pressure p_1, expands to a pressure p_2, and then exhausts to a condenser in which the pressure is one pound—that is, the vacuum is 28 inches, which is the case of the ordinary reciprocating condensing engine. Table VI. gives the output when exhausting to the atmosphere, as in the case of locomotives and other engines which have no condenser. It will be seen that a turbine supplied with steam at 100 lbs. pressure (absolute) can do the same work as a triple compound reciprocator expanding steam of 150 lbs. to 5 lbs., or one of 200 lbs. expanding to 7 lbs. And a locomotive would require 55 per cent. more steam at 200 lbs. pressure to do the same work, supposing the efficiency of the means is the same in each case. It will also be seen that steam at 100 lbs. has a 19 per cent. higher potential than that of 50 lbs; that 150 is 8·44 per cent. higher than 100; that 200 is 5·77 per cent. higher than 150; and 250 is 4·82 per cent. higher than 200. Also that the 250 lbs. of the new quadruple has a potential 7·41 per cent. greater than the 175 lbs. of the older triples.

Table V.—Maximum Work possible from One Pound of Steam of Pressure p_1, Expanding Adiabatically to Pressure p_2, and Exhausting to Condenser at 1 lb. Pressure.

Initial pressure.	100 lbs.		125 lbs.		150 lbs.		175 lbs.		200 lbs.		225 lbs.		250 lbs.	
Termina p_2	B.T.U.	H.P.	B.T.U.	H.P.	B.T.U.	H.P.	B.T.U.	H.P.	B.T.U.	H.P.	B.T.U.	H.P.	B.T.U.	H.P.
1·0 lb.	288·0	6·80	303·0	7·14	314·5	7·42	324·0	7·64	332·5	7·84	340·6	8·03	348·0	8·21
2·0 lbs.	286·8	6·76	301·8	7·09	313·3	7·26	322·4	7·61	331·3	7·81	339·3	8·00	346·8	8·18
3·0 ,,	277·2	6·54	292·6	6·90	309·2	7·17	313·5	7·40	322·2	7·60	330·2	7·80	337·2	7·95
4·0 ,,	266·2	6·28	281·9	6·65	294·0	6·93	303·2	7·15	312·0	7·36	320·2	7·55	328·2	7·74
5·0 ,,	258·0	6·09	273·5	6·45	285·0	6·72	295·0	6·96	304·0	7·17	312·0	7·36	320·0	7 55
6·0 ,,	249·9	5·90	266·9	6·27	278·0	6·55	287·9	6·80	297·4	7·01	306·8	7·23	313·9	7·40
7·0 ,,	243·9	5·75	259·3	6·11	271·9	6·42	281·8	6·64	290·8	6·86	300·3	7·09	307·8	7·26
8·0 ,,	237·5	5·61	253·5	5·96	265·5	6·26	276·8	6·53	285·5	6·73	294·5	6·94	302·5	7·13
9·0 ,,	232·0	5·47	248·0	5·85	260·5	6·14	271·3	6·40	280·7	6·62	289·1	6·82	297·0	7·00
10 ,,	226·9	5·35	242·1	5·71	254·0	6·00	265·5	6·26	275·9	6·51	284·1	6·70	292·1	6·89
11 ,,	222·1	5·24	237·1	5·59	249·8	5·89	260·7	6·15	270·5	6·38	279·1	6·58	287·1	6·77
12 ,,	217·0	5·12	233·0	5·49	245·5	5·80	256·1	6·04	265·4	6·26	275·0	6·48	283·0	6 67
13 ,,	212·2	5·00	228·6	5·39	240·7	5·69	251·6	5·93	261·1	6·16	270·7	6·39	278·7	6·57
14 ,,	207·4	4·90	224·1	5·28	236·4	5·60	247·2	5·83	257·3	6·07	266·4	6·29	274·4	6·47
15 ,,	204·0	4·81	220·0	5·19	232·0	5·47	243·5	5·74	253·0	5·97	262·0	6·18	270·0	6·37
16 ,,	200·6	4·73	216·6	5·11	228·6	5·39	239·6	5·64	249·9	5·89	258·8	6·10	266·4	6·28
17 ,,	197·4	4·65	213·3	5·03	225·0	5·30	236·0	5·56	246·5	5·81	255·3	6·02	262·6	6·19
18 ,,	194·3	4·58	210·1	4·95	220·0	5·23	232·9	5·49	243·3	5·74	252·0	5·94	259·9	6·11
19 ,,	191·3	4·50	207·2	4·88	219·4	5·17	230·0	5·42	240·5	5·67	249·3	5·88	257·3	6·06
20 ,,	188·4	4·44	204·4	4·82	217·3	5·12	228·0	5·38	238·0	5·61	247·0	5·82	255·4	6·02
25 ,,	176·0	4·15	191·3	4·51	205·0	4·83	215·9	5·09	225·4	5·31	234·2	5·52	243·0	5·73
30 ,,	164·8	3·89	180·0	4·24	194·5	4·59	205·0	4·83	214·9	5·07	223·4	5·27	232·0	5·47
35 ,,	155·2	3·66	169·6	4·00	184·2	4·34	195·2	4·60	205·2	4·84	214·2	5·05	222·2	5·24
40 ,,	146·2	3·45	160·5	3·80	175·1	4·13	186·0	4·40	196·0	4·62	205·0	4·81	213·0	5·02

Table VI.—Maximum Work possible from One Pound of Steam of Pressure p_1, Expanding Adiabatically to Pressure p_2, and Exhausting against 15 lbs.

Initial pressure.	p_1 100 lbs.		p_1 125 lbs.		p_1 150 lbs.		p_1 175 lbs.		p_1 200 lbs.		p_1 225 lbs.		p_1 250 lbs.	
Terminal p_2.	B.T.U.	H.P.	B.T.U.	H.P.	B.T.U.	H.P.	B.T.U.	H.P.	BT.U.	H.P.	B.T.U.	H.P.	B.T.U.	H.P.
15 lbs.	137·0	3·23	153·1	3·61	165·0	3·89	176·5	4·16	186·5	4·40	195·4	4·61	203·3	4·80
20 ,,	135·7	3·20	152·3	3·60	164·3	3·88	176·0	4·15	186·1	4·39	195·1	4·60	203·1	4·79
25 ,,	132·6	3·13	149·6	3·53	162·6	3·83	173·6	4·09	183·6	4·33	192·6	4·54	201 6	4·75
30 ,,	128·7	3·04	145·4	3·43	158·4	3·74	169·4	4·00	179·4	4·23	189·4	4·47	197·5	4 66
35 ,,	123·8	2·92	140·6	3·32	153·8	3·63	165·1	3·90	175·1	4·13	184·4	4·35	192·1	4·53
40 ,,	119·6	2·82	135·6	3·20	148·6	3·50	160·6	3·79	170·6	4·02	179·6	4·23	187·6	4·42
45 ,,	114·5	2·70	130·0	3·07	143·5	3·40	155·0	3·66	165·0	3·89	174·0	4·10	182·0	4·30
50 ,,	108·0	2·55	124·4	2·93	138·8	3·27	150·6	3·54	160·5	3·78	169·0	3·99	177·0	4·18
55 ,,	103·2	2·43	119·7	2·82	133·9	3·16	145·4	3·43	155·4	3·67	164·4	3·88	172·4	4·07
60 ,,	98·2	2·32	114·7	2·71	130·4	3·08	141·9	3·35	151·4	3·57	160·4	3·78	168·4	3·97
65 ,,	93·4	2·20	110·2	2·61	127·2	3·00	138·2	3·26	147·7	3·48	155·6	3·67	164·2	3·88
70 ,,	88·7	2·09	106·8	2·52	123·8	2·92	134·8	3·18	144·1	3·40	152·1	3·59	160·1	3·78
75 ,,	84·2	1·99	104·6	2·46	120·0	2·83	130·4	3·08	140·3	3·31	148·2	3·50	156·2	3·69
80 ,,	79·9	1·89	101·2	2·38	114·8	2·71	126·5	2·99	136·6	3·22	144·9	3·42	151·7	3·59
85 ,,	76·3	1·80	98·1	2·31	110·9	2·61	122·9	2·90	133·3	3·14	141·7	3·34	148·7	3·50
90 ,,	72·9	1·72	94·8	2·24	108·4	2·56	119·9	2·83	129·5	3·06	138·3	3·26	145·3	3·42
95 ,,	70·2	1·65	91·4	2·16	104·4	2·46	116·9	2·76	126·5	2·98	135·3	3·19	141·7	3·34
100 ,,	68·3	1·61	88·3	2·09	102·3	2·40	113·3	2·67	123·3	2·90	131·9	3·11	138·4	3·27

Table VII.—Maximum Work possible from One Pound of Steam of Pressure p_1, Expanding Adiabatically to Pressure p_2, and Exhausting against Pressure p_2.

p_1.	25 lbs.		50 lbs.		75 lbs.		100 lbs.		125 lbs.		150 lbs.		175 lbs.		200 lbs.		225 lbs.		250 lbs.	
p_2.	B.T.U.	H.P.	B.T.U.	H.P.	B.T.U.	H.P.	B.T.U.	H.P.	B.T.U.	H.P.	B.T.U.	H.P.	B.T.U.	H.P.	B.T.U.	H.P.	B.T.U.	H.P.	B.T.U.	H.P.
1·0	188	4·43	243	5·73	269	6·34	289	6·81	303	7·14	314	7·41	324	7·64	332	7·83	340	8·02	348	8·21
2·0	163	3·85	209	4·93	234	5·52	254	6·00	269	6·34	282	6·64	292	6·88	300	7·08	307	7·24	315	7·43
3·0	140	3·30	187	4·41	213	5·02	233	5·50	248	5·85	260	6·12	270	6·39	278	6·56	286	6·74	293	6·90
4·0	121	2·85	169	4·00	196	4·62	216	5·10	231	5·45	243	5·72	253	6·00	262	6·18	270	6·37	278	6·56
5·0	107	2·52	156	3·68	184	4·34	204	4·81	219	5·16	231	5·45	241	5·69	250	5·90	258	6·09	266	6·27
6·0	97·1	2·29	146	3·44	174	4·10	193	4·55	210	4·94	222	5·24	232	5·47	241	5·67	250	5·90	257	6·06
7·0	88·2	2·08	137	3·22	165	3·89	185	4·36	201	4·73	213	5·02	223	5·26	232	5·47	242	5·70	249	5·87
8·0	80·2	1·89	128	3·00	157	3·70	177	4·17	193	4·55	206	4·85	216	5·10	225	5·31	234	5·52	242	5·71
9·0	72·8	1·72	121	2·84	150	3·54	170	4·00	186	4·39	199	4·69	209	4·94	219	5·16	228	5·37	235	5·54
10	65·8	1·55	114	2·70	144	3·40	164	3·87	179	4·22	192	4·52	202	4·77	213	5·01	221	5·21	229	5·40
11	59·2	1·40	108	2·55	138	3·27	158	3·73	173	4·08	186	4·39	196	4·62	206	4·87	215	5·07	223	5·26
12	53·0	1·23	103	2·43	132	3·11	152	3·56	168	3·96	181	4·26	190	4·48	200	4·73	210	4·95	218	5·12
13	47·3	1·12	97·2	2·29	126	2·97	147	3·46	163	3·85	175	4·13	185	4·36	195	4·61	205	4·83	213	5·02
14	41·9	0·99	92·3	2·16	121	2·85	141	3·32	158	3·73	170	4·01	181	4·27	191	4·50	200	4·72	208	4·90
15	36·9	0·87	87·7	2·07	117	2·75	137	3·23	153	3·61	165	3·89	177	4·16	186	4·40	195	4·60	203	4·79
20	14·6	0·34	68·2	1·61	97·7	2·30	118	2·80	135	3·19	147	3·44	159	3·75	169	4·00	178	4·20	186	4·40
25	..	..	52·0	1·23	81·9	1·94	103	2·43	120	2·83	134	3·16	144	3·40	154	3·63	163	3·85	172	4·06
30	..	..	38·2	0·90	68·8	1·62	91·3	2·15	108	2·55	121	2·85	132	3·11	142	3·35	152	3·59	160	3·78
35	..	..	27·0	0·64	58·6	1·33	80·7	1·90	97·5	2·30	111	2·62	122	2·88	132	3·11	141	3·32	149	3·51
40	..	..	16·8	0·40	49·2	1·16	72·0	1·70	88·0	2·08	101	2·39	113	2·66	123	2·90	132	3·11	140	3·30
45	..	..	8·1	0·19	41·0	0·97	63·5	1·50	79·0	1·86	92·2	2·18	104	2·45	114	2·69	123	2·90	131	3·03
50	..	..	..	..	32·7	0·77	54·0	1·27	70·4	1·66	84·8	2·00	96·6	2·28	107	2·52	115	2·71	123	2·90
55	..	..	..	..	25·3	0·60	46·8	1·10	63·3	1·49	77·5	1·83	89·0	2·10	99·0	2·33	108	2·55	116	2·74
60	..	..	..	..	18·2	0·43	39·8	0·94	56·8	1·33	72·0	1·70	83·5	1·97	93·0	2·19	102	2·40	110	2·60
65	..	..	..	..	12·0	0·28	33·2	0·80	51·4	1·18	67·0	1·59	78·0	1·84	87·5	2·06	95·4	2·25	104	2 45
70	..	..	..	..	6·0	0·14	26·9	0·63	45·4	1·07	62·0	1·46	73·0	1·72	82·3	1·94	90·3	2·08	98·3	2·32
75	..	..	..	..	..	..	21·0	0·50	41·5	0·97	56·8	1·32	67·2	1·58	77·1	1·82	85·0	2·00	93·6	2·21
80	..	..	..	..	..	..	15·5	0·37	36·8	0·87	50·4	1·18	62·1	1·47	72·2	1·70	80·5	1·90	87·3	2·06
85	..	..	..	..	..	..	10·8	0·25	32·6	0·77	45·4	1·07	57·4	1·35	67·8	1·59	76·2	1·80	83 2	1·96
90	..	..	..	..	..	..	6·48	0·15	28·3	0·67	41·9	0·99	53·4	1·25	63·0	1·49	71 8	1·70	78·8	1·86
95	..	..	..	..	..	..	2·79	0·06	24·0	0·57	37·3	0·88	49·5	1·17	59·1	1·40	67·9	1·60	74·8	1·76
100	..	..	..	..	..	..	..	..	20·0	0·47	34·0	0·80	45·2	1·07						

Efficiency as affected by Jacketing.

Careful investigations show that all types of steam engines are rendered more efficient by the addition of steam jackets, and that the more completely the hot surfaces of cylinders, receivers, &c., are jacketed, the greater is the saving effected.

This amounts to saying, that for every pound of steam condensed in the jackets, some greater quantity is saved in the cylinders. The ratio of steam saved in the cylinders to steam expended in the jackets, varies from a little under 2 to 1 in some types of engine, to over 5 to 1 in other types.

The gain that may be expected to result from jacketing an engine depends on such a multitude of considerations—relating not only to the design of the engine and of the boiler, but also to the management of the machinery under steam—that it can only be very generally stated as lying between 5 and 25 per cent. of the total feed-water evaporated; but in the case of modern marine machinery, of good construction, it is not probable that the gain is over 10 per cent.

The limit of usefulness of jacketing is reached when the exhaust is just free from particles of water in suspension.

Small cylinders are more benefited by jacketing than large ones, in consequence of the ratio of area of hot surface to cubic contents being greater than in large ones; and slow moving engines benefit more than quick.

It is important that there should be a thorough circulation of steam in the jackets, but the plan of passing the steam through the jackets on its way to the H.P. cylinder should be avoided.

THE RESISTANCE AND PROPULSION OF SHIPS.

In dealing with cubes, or with parallelepipeds of similar form, immersed in water until the uppermost face is just flush with the surface, it is found, on making the necessary calculations, that the wetted surface is exactly proportional to the $\frac{2}{3}$rd power (or the square of the cube root) of the displacement. Taking the case of cubes,—

Let L = length of edge,
D = displacement,
W = wetted surface;

then $D = L^3$, or $L = \sqrt[3]{D}$,

and $W = 5 \times L^2 = 5 \times (\sqrt[3]{D})^2$.

That is, W varies as $D^{\frac{2}{3}}$.

It may also be noted that $L^2 = (\sqrt[3]{D})^2$,—that is, L^2 (which corresponds to mid-ship section) also varies as $D^{\frac{2}{3}}$; and therefore W varies as L^2, or, in other words, wetted surface varies as area of mid-ship section.

These results are not quite accurate for parallelepipeds which are not similar in form (that is, whose lengths, breadths, and depths are not of the same relative proportions), but the inaccuracy is only slight within practicable limits; so that, if ships of ordinary proportions are

substituted for cubes or parallelepipeds, it is practically correct to sa that wetted surface varies as area of mid-ship section, and also as $\frac{2}{3}$r power of displacement.

For similar vessels displacement is also a measure of the fineness o the lines, since, when length and mid-ship section are the same, i varies directly as the prismatic co-efficient of fineness.

Now the resistances of ships depend almost entirely upon these tw elements,—wetted surface, and form, or fineness of lines, and may be classed under the four heads—

(1) Resistance due to skin friction ;
(2) Resistance due to eddy making ;
(3) Resistance due to wave making ;
(4) Resistance (augmented) due to the action of the propeller on the ship.

The first of these depends on the extent and nature of the wetted surface, and the depth of immersion ; the third, on the lines of the ship, and on her degree of fineness ; and the second, on all of these combined. The fourth arises from the displacement of the water and the consequent reduction of "head" and pressure on the stern. Propellers of large diameter and fine pitch produce much *augmented resistance* as well as negative slip due to the big *wake currents*.

Residual Resistance—that is, the resistance from all other causes than skin friction can be estimated with a fair degree of accuracy by the following formula of Mr. D. W. Taylor, U.S.A. :—

Rule 21. Residual resistance in lbs. $= \frac{12{\cdot}5\ b\mathrm{DV}^4}{\mathrm{L}^2}$

b is the *block* co-efficient ; D, the displacement in tons ; V, the speed in knots ; L, the length on waterline in feet.

This formula is applicable only to speeds for which $\frac{\mathrm{V}^2}{\mathrm{L}}$ is less than 1·2.

For merchant ships, with block co-efficients 0·6 to 0·65, this is a good formula for speeds, such that $\frac{\mathrm{V}^2}{\mathrm{L}} = 0{\cdot}85$; when co-efficients are 0·5 to 0·55, then $\frac{\mathrm{V}^2}{\mathrm{L}} = 1$.

It is therefore evident, from the above considerations, that the old speed and power formulæ rest on a sound basis, and are capable, in careful hands, of giving fairly accurate results. These formulæ are :—

Rule 22. $$\text{I.H.P.} = \frac{\mathrm{D}^{\frac{2}{3}} \times \mathrm{S}^3}{\mathrm{C}}$$

and $$\text{I.H.P.} = \frac{\text{area of immersed mid-ship section} \times \mathrm{S}^3}{\mathrm{K}}$$

where **D**, is displacement in tons ; S, speed in knots ; and C and K co-efficients.

The peculiar value of these formulæ lies in the fact that they can be applied at a very early stage of the work, before such data as angles of

obliquity of stream lines can be obtained with any degree of accuracy: and thus the power, approximate weight, and an outline drawing of the machinery can be got out simultaneously with the design of the vessel,—a great advantage where time is limited, as is usually the case in preparing tenders.

It is important to notice, in connection with the above formulæ, that —although the resistance of a ship, moving uniformly at any speed, may vary as the *square* of that speed,—the power required to overcome the resistance, and propel it at any speed, varies as the *cube* of that speed. For,—

Let S = speed in feet per minute;
R = resistance in pounds at that speed;
Then $R = S^2 \times C$, —where C is a co-efficient;
and, multiplying both sides by S

$$R \times S = S^3 \times C.$$

But $R \times S$ is the work done, in foot-pounds per minute, in overcoming the resistance R through the space S, and, divided by 33,000, is equal to the nett horse-power required to drive the ship. This law only holds for similar ships driven at corresponding speeds (speeds proportional to the square roots of the linear dimensions), since it is evident that the resistance due to wave making can only be proportional under these conditions.

The second formula is useful as a check, or corrective to the first, where the vessels under consideration are not absolutely similar, but have same ratio of length to breadth and draught, with a variation in the rise of floor.

Table VIII. (given by Sir W. H. White) shows the values of C for some typical ships of very different classes, at various speeds, and whilst indicating generally the range of the variations that occur, serves also to show the difficulties that the naval architect has to encounter in obtaining high speeds in vessels of small dimensions.

It is, perhaps, scarcely necessary to add that these co-efficients of performance represent the *combined* efficiency of ships and machinery, and that they are therefore just as liable to be effected by an unsuitable propeller as by a foul bottom or unsuitable lines; and also that for every model there is, as a rule, only one speed of maximum efficiency, though it is evident from the type of curve usually obtained, that there may be two speeds at which the efficiency is equal and slightly below the maximum.

It should also be borne in mind that accurately determined co-efficients of performance, &c., generally apply to more or less smooth-water conditions, and that a form of vessel which gives highest speed with least power, on such trials, may be far from the best for an ocean-going steamer.

As a rule, length assists speed especially in a sea-way.

The gain in economy of propulsion resulting from increase in dimensions is made very clear by the following figures, which are derived from the trials of certain cruisers:—

Table VIII.—Relation of Powers and Speeds.

Description of Vessel.	Length in feet.	Breadth in feet.	Draught (mean). ft. in.	Displacement in tons.	I.H.P's. and Admiralty Co-efficients.							
					10 Knots.		14 Knots.		18 Knots.		20 Knots.	
					I.H.P.	Co-eff.	I.H.P.	Co-eff.	I.H.P.	Co-eff.	I.H.P.	Co-eff.
Torpedo Boat,	135	14	5 1	103	110	200	260	232	870	147	1,130	156
Torpedo Gunboat, "Sharpshooter" Class,	230	27	8 3	735	450	181	1100	203	2,500	190	3,500	186
3rd Class Cruiser, "Medusa,"	265	41	16 6	2,800	700	284	2100	259	6,400	181	10,000	159
2nd Class Cruiser, "Terpsichore,"	300	43	16 2	3,330	800	279	2400	255	6,000	217	9,000	198
1st Class Cruiser, "Edgar,"	360	60	23 9	7,390	1000	380	3000	347	7,500	295	11,000	276
1st Class Cruiser, "Blenheim,"	375	65	25 9	9,100	1500	290	4000	298	9,000	282	12,500	278
Atlantic Passenger Steamer,	525	63	21 3	11,550	2000	255	4600	304	10,000	297	14,500	281

Table VIIIa.—Relation of Powers and Displacements.*

	No. 1.	No. 2.	No. 3.	No. 4.	No. 5.
Length in ft. . . .	280	300	360	435	500
Breadth in ft. . .	35	43	60	69	71
Mean draught in ft. .	13	16½	23¾	24½	26½
Displacement in tons .	1800	3400	7400	11000	14200
I.H.P. for 20 knots .	6000	9000	11000	14000	15500
I.H.P. per ton of displacement . . .	3·3	2·65	1·48	1·27	1·09

The following horse-powers were required to drive cruisers Nos. 4 and 5 in the above Table at the speeds named :—

	No. 4.	No. 5.
10 knots	1500 I.H.P.	1800 I.H.P.
12 ,,	2500 ,,	3100 ,,
14 ,,	4000 ,,	5000 ,,
16 ,,	6000 ,,	7500 ,,
18 ,,	9000 ,,	11000 ,,
20 ,,	14000 ,,	15500 ,,
22 ,,	23000 ,,	23000 ,,

The frictional resistance of clean painted surfaces varies about as the 1·83 power of the speed (compare Table XVI.), but resistance due to wave making may vary very widely, since it is dependent on form. The total resistance of "Destroyers" has been found to vary as follows* :—

Up to 11 knots, ...	nearly as speed ^{2}
At 16 ,, ...	,, speed ^{3}
,, 18-20 ,, ...	,, speed $^{3\cdot3}$
,, 22 ,, ...	,, speed $^{2\cdot7}$
,, 25 ,, ...	,, speed ^{2}
,, 25-30 ,, . . .	practically as speed $^{1\cdot83}$

and the resistances other than frictional vary as follows :—

Up to 11 knots	as speed ^{2}
At 12½ to 13 knots	,, speed ^{3}
,, 14½ knots	,, speed ^{4}
,, 18 ,,	,, speed (more than 5th power).
,, 24 ,,	,, speed ^{2}

and at higher speeds as still lower powers of the speeds.

* Sir Wm. White, British Association Address, 1899.

The relation of the frictional resistance to the total resistance at various speeds is computed to be as follows* :—

	"Destroyer."	Cruiser.
At 12 knots	80 per cent.	90 per cent.
,, 16 ,,	70 ,,	85 ,,
,, 20 ,,	nearly 50 ,,	nearly 80 ,,
,, 23 ,,	...	over 70 ,,
,, 30 ,,	45 ,,	...

And it may be remarked that, if the coefficient of friction be doubled (as Table XVI. shows it might easily be with a foul bottom) the maximum speed of the "Destroyer" would fall fully 5 knots, and that of the cruiser would be reduced from 23 knots to 19.

Recent "Destroyers" for the British Navy have been of following dimensions, etc. :—

	Ft. long.	Tons displ.	I.H.P.	Knots.
1893	180	240	4000	26–27
1896	200–210	280– 300	5500–6000	30
1899	230	360– 380	8000–10,000	32–33
1903	225–230	550– 590	7000	25·5
1907	250–270	860– 960	14,000	33
1910	270–280	900–1000	12,000–16,000	27–33·5

The propelling apparatus of these boats has given, on an average, 45 I.H.P. per ton of weight. In the 30 knot boats nearly 50 per cent. of the displacement has been allotted to the propelling apparatus, and the load of fuel and equipment has taken 12 to 14 per cent. more.

* Sir Wm. White, British Association Address, 1899.

Table IX.—Results of Trials of Steamships.

Description.	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
Length, perpendiculars, ft.,	630·0	330·0	290·0	270·0	630·0	440·0	220·0	530·0	400·0	340·0	470·0	320·0	250·0
Breadth, extreme, ft.,	68·0	40·0	32·0	32·0	70·0	52·0	32·0	58·0	46·2	32·0	52·0	40·0	32·0
Draught of water, mean, ft.,	29·9	20·1	11·9	8·8	30·0	20·1	9·0	22·5	16·6	13·4	24·6	16·2	12·2
Displacement (tons),	25,910	6110	2190	1257	30,020	8156	736	14,290	5942	2941	12,380	4729	1740
Midship section immersed, area,	1925	1050	414	286	2080	980	206	1225	747	455	1310	686	256
Wetted skin, sq. ft.,	72,990	25,540	12,170	9075	76,440	25,610	6841	46,390	25,965	12,022	42,380	21,102	11,184
Length of fore-body,	212·0	156·4	113·2	106·3	178·1	170·1	96·0	134·0	120·2	87·2	98·0	72·0	70·0
Angle of entrance,	17° 4′	15° 36′	17° 32′	16° 6′	22° 11′	15° 10′	14° 12′	22° 4′	18° 2′	27° 0′	20° 20′	22° 28′	21° 36′
$\frac{\text{Displacement}}{\text{Length} \times \text{mid. section}}$	·725	·611	·580	·502	·787	·711	·568	·703	·711	·644	·702	·724	·676
$\frac{\text{Displacement} \times 35}{\text{Length} \times \text{mid. section}}$	·650	·561	·609	·805	·725	·622	·572	·703	·644	·801	·739	·771	·646
Speed (knots),	20·5	21·09	20·24	18·9	19·20	18·52	10·5	14·7	14·79	12·22	12·69	12·79	12·44
Ind. horse-power,	20,500	10,660	5500	3230	11,400	6347	1400	6116	3741	1586	3789	1762	802
I.H.P. per 10 feet wetted skin,	28·1	42·1	44·2	35·6	16·23	25·29	20·46	13·17	14·39	13·2	8·71	8·47	6·9
Ind. reduced to 10 knots,	4·05	4·52	4·27	4·46	2·85	4·44	4·56	4·75	4·44	4·2	4·96	4·17	4·03
$\frac{\text{Displ.} \times \text{S}^2}{\text{I.H.P.}}$	257	255	229	246	253	257	201	205	289	244	236	255	235
$\frac{\text{Immersed mid. sec.} \times \text{S}^3}{\text{I.H.P.}}$	652	615	580	556	712	698	656	657	845	685	601	610	657
I.H.P. ÷ displacement,	1·02	1·75	1·77	2·01	0·42	0·78	1·90	0·42	0·65	0·46	0·27	0·23	0·55

Ratios.

Table X.—Results of Trials

Particulars.	QSS LCNA	TSS KWO	TSS OCNC	TSS DLD	TSS CPA	TSS CRMA	HMS INBL	HMS DNT	HMS GLS
Length, perpdlrs, ft.,	760	678	685	668	600	678	530	490	430
Breadth, extreme, ft.,	87·5	72·0	68·0	67·0	65·2	72·4	78·5	82·0	47·0
Drft water, mean, ft.,	32·5	29·5	29·9	29·0	29·0	30·0	26·0	26·5	15·25
Displacement (tons),	36,440	26,500	25,910	23,620	21,500	27,500	17,250	17,900	4,800
Imrsd mid sectn, sq.ft.	2,500	1,950	1,922	1,753	1,720	2,064	1,868	1,912	624
Wetted skin, sq. ft.,	82,350	65,150	67,209	62,110	54,780	68,700	47,340	44,630	22,250
Prism co-efficient, .	0·671	0·713	0·690	0·715	0·729	0·679	0·660	0·669	0·625
$0{\cdot}4\sqrt[4]{L}+\sqrt[3]{S}$., . .	0·714	0·712	0·725	0·701	0·704	0·748	0·637	0·680	0·613
Speed (knots), . .	25·40	23·50	20·80	23 50	22·09	20·19	27·36	21·25	26·30
Number of screws, .	4	2	2	2	2	3	4	4	4
Engines, . . .	Turbs	Recip	Recip	Recip	Recip	Turbs	Turbs	Turbs	Turbs
I.H.P. or S.H.P., .	64,600	39,000	26,500	36,000	29,936	24,000	47,300	24,712	25,417
H.P. per 100 sq. ft. wetted skin.	78·4	59·90	39·4	57·9	54·8	35·00	100·0	55·4	114
Do. redcd to 10 kns.	4·7	4·62	4·38	4·47	5·08	4·25	4·88	5·78	6·27
$D^{\frac{2}{3}} \times S^3 \div H.P.$ . .	278	297	297	297	278	312	289	264	203
$H.P. \div Dispmt^{\frac{2}{3}}$ . .	58·7	43·7	30·2	44·9	38·8	26·3	70·8	36·1	89·2
Service of ship, . .	Atl Ex	Atl Ex	Atl Ex	Atl Ex	Atl Ex	Atl Ex	Cruiser	Bttlsp	Cruiser

Various Ships of High Speed.

HMS BLN	HMS SFT	TSS BMC	HMS SKR	PSS EMQ	PSS LMG	TSS QN	TSS PG	TSS SNA	TSS NRMA	PSS LND	PSS RS
385	345	375	360	360	330	310	290	284	290	338	310
41·5	34·2	46·0	40·0	42·3	40·0	40·0	38·0	39·0	36·0	34·75	33 0
13·5	10·5	13·4	14·25	13·0	8·75	10·5	11·92	12·0	12·0	9·00	6·0
3,360	1,800	3,353	2,945	2,940	1,868	2,080	2,100	1,990	1,875	1,700	1,063
476	301	590	500	466	322	375	416	420	390	292	188
18,100	12,000	18,830	16,350	16,190	13,082	12,200	13,170	11,950	11,700	11,800	9,920
0·640	0·609	0·530	0·572	0·623	0·615	0·625	0·609	0·554	0·580	0·607	0·635
0·585	0·524	0·617	0·594	0·620	0·604	0·602	0·605	0·608	0·612	0·617	0·623
27·8	36·00	23·27	25·20	21·71	22·3	21·73	20·34	20·00	19·70	21·50	19·6
4	4	3	2	Paddle	Paddle	3	2	3	2	Paddle	Pdle
Turbs	Turbs	Turbs	Recip	3 Cyl Comp	Diag Comp	Turbs	4 Ck Trpl	Turbs	Geared Turbs	Comp	Diag Comp
18,000	30,000	14,700	16,899	11,442	7,500	8,000	5,820	6,670	5,000	7,000	3,500
99·4	250	78·1	103·4	70·6	57·3	65·6	44·2	55·8	42·8	50·3	35·3
4·63	5·35	6·17	6·46	6·96	5·16	6·19	5·27	6·97	5·60	5·97	4·69
267	230	192	202	183	223	210	236	190	232	202	223
80·4	203	65·6	82·2	55·8	49·3	49·1	..	42·2	32·9	49·3	33·6
Cruiser	Scout	Home Ex	Scout	Home Ex	Home Ex	Home Ex	Open S Ex	Home Ex	Home Ex	Ch'nn'l	Home Ex

Kirk's Analysis.

The following very simple and useful method of estimating wetted skin and comparing forms of ships is generally known as "Kirk's analysis":—

A diagram, resembling fig. 5, called the "block model" is first made, the dimensions being determined as follows:—

Length AB = length of ship (from forward side of stem to after side of stern-post, at mean trial draught).

Depth KL = depth of ship from mean trial draught to top of keel (if any).

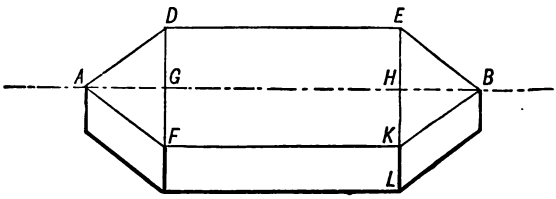


FIG. 5.

$$\text{Breadth EK} = \frac{\text{Area of immersed mid-ship section}}{\text{KL}}.$$

$$\text{AH} = \text{GB} = \frac{\text{Displacement in tons} \times 35^*}{\text{Area of immersed mid-ship section}}.$$

$$\text{Length of AD} = \sqrt{\text{AG}^2 + \text{GD}^2}.$$

Then, the wetted surface of the "block model" is,—

$$(\text{EK} \times \text{AH}) + (2\text{KL} \times \text{FK}) + (4\text{KL} \times \text{AD}).$$

The angle of entrance is EBK; EBH is half that angle; and tan. $\text{EBH} = \frac{\text{EH}}{\text{HB}}$; or, the tangent of half the angle of entrance is equal to $\frac{\text{Half breadth of model}}{\text{Length of fore-body}}$, and from this, by means of a table of natural tangents, the angle of entrance may be obtained.

* 35 cubic feet of salt water weigh one ton, and are therefore equal to one ton displacement.

Table XI.—Angles of Entrance, given in degrees, suitable for various Speeds and Lengths.

Length of Ship.	Speeds of Ship in Knots.															
	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25
feet																
200	26·3	23·9	21·9	20·1	18·8	17·6	16·5	15·5	14·6	13·9	13·2	12·5	12·0	11·4	11·0	10·5
250	27·8	25·3	23·2	21·4	19·9	18·6	17·4	16·4	15·5	14·6	13·9	13·3	12·7	12·1	11·6	11·1
300	29·1	26·5	24·3	22·4	20·8	19·4	18·2	17·1	16·2	15·3	14·6	13·9	13·2	12·7	12·1	11·7
350	30·3	27·5	25·2	23·3	21·6	20·2	18 9	17·8	16·8	15·9	15·1	14·4	13·8	13·2	12·6	12·1
400	31·3	28·5	26·1	24·1	22·4	20·9	19·6	18·4	17·4	16·5	15·7	15·0	14·2	13·6	13·1	12·5
450	32·2	29·3	26·9	24·8	23·2	21·5	20·2	19·0	17·9	17·0	16·1	15·4	14·7	14·0	13·4	12·9
500	33·1	30·1	27·6	25·5	23·7	22·1	20·7	19·5	18·4	17·4	16·6	15·8	15·1	14·4	13·8	13·3
550	33·9	30·8	28·2	26·1	24·2	22·6	21·2	19·9	18·8	17·8	17·0	16·1	15·4	14·7	14·1	13·6
600	34·7	31·5	28·9	26·7	24·8	23·1	21·7	20·4	19·3	18·3	17·3	16·5	15·8	15·1	14·4	13·9
650	35·4	32·1	29·4	27·2	25·2	23·6	22·1	20·8	19·6	18·6	17·7	16·8	16·1	15·4	14·7	14·1
700	36·0	32·7	30·0	27·7	25·7	24·0	22·5	21·2	20·0	19·0	18·0	17·1	16·4	15·7	15·0	14·4
750	36·6	33·3	30·5	28·2	26·2	24·4	22·9	21·6	20·4	19·3	18·3	17·5	16·7	15·9	15·3	14·7
800	37·2	33·8	31·0	28·6	26·6	24·8	23·2	21·9	20·7	19·6	18·6	17·7	16·9	16·2	15·5	14·9

If ships are of very fine form for the speed required, and assuming that I.H.P. varies at S^3 the power per 100 feet of wetted skin at 10 knots should be 4·0 I.H.P. ; on the other hand, if the ship is not sufficiently fine for her maximum speed, the efficiency will be much less and the power at 10 knots may be then 5·0. That is, the power required to drive a ship of fairly good lines at a speed S knots is,—

Rule 22a. I.H.P. per 100 square feet W.S. $= \frac{S^3 \times 4{\cdot}5}{10^3} = S^3 \times 0{\cdot}0045.$

Example:—To find the I.H.P. necessary to drive a ship at 15 knots, the wetted skin of "block-model" being 16,200 square feet, the lines rather full.

The I.H.P. per 100 square feet $= 15^3 \times 0{\cdot}005 = 16{\cdot}875$

and I.H.P. required $= 16{\cdot}875 \times 162 = 2744$.

In ordinary practice the wetted surface of the "block model" is found to exceed that of the actual ship by 2 per cent. (in the case of full ships), by 3 to 5 per cent. for ordinary steamers, and as much as 8 per cent. in the case of very fine steamers.

The following Table gives, on inspection, the horse-powers required per 100 feet of wetted surface at various speeds and rates, and will facilitate calculations by the above method :—

Table XII.—I.H.P. per 100 feet of wetted surface at different speeds.

Speeds in Knots.	I.H.P. per 100 sq. feet of wetted surface at 10 knots.												
	4·0	4·1	4·2	4·3	4·4	4·5	4·6	4·7	4·8	4·9	5·0	5·1	5·2
9	2·92	2·99	3·06	3·14	3·21	3·28	3·35	3·42	3·49	3·57	3·64	3·71	3·78
9·5	3·43	3·51	3·60	3·68	3·77	3·85	3·94	4·02	4·11	4·19	4·28	4·36	4 45
10	4·00	4·10	4·20	4·30	4·40	4·50	4·60	4·70	4·80	4·90	5·00	5·10	5 20
10·5	4·63	4·75	4·86	4·98	5·09	5·21	5·32	5·44	5·56	5·79	5·79	5·90	6·02
11	5·32	5·46	5·59	5·72	5·86	5·99	6·12	6·25	6·39	6·52	6·65	6·79	6·92
11·5	6·08	6·23	6·39	6·54	6·69	6·84	6·99	7·15	7·30	7·45	7·60	7·76	7·91
12	6·91	7·08	7·26	7·43	7·60	7·78	7·95	8·12	8·29	8·47	8·64	8·81	8 99
12 5	7·81	8·01	8·20	8·40	8·59	8·79	8·98	9·18	9·37	9·57	9·76	9·96	10·15
13	8·79	9·01	9·23	9·45	9·67	9·89	10·10	10·32	10·54	10·76	10·98	11·20	11 42
13·5	9·84	10·09	10·33	10·58	10·82	11·07	11·32	11·56	11·81	12·05	12·30	12·55	12·79
14	10·98	11·25	11·52	11·80	12·07	12·35	12·62	12·90	13·17	13·44	13·72	13·99	14·27
14·5	12·19	12·50	12·80	13·11	13·41	13·72	14·02	14·33	14·63	14·94	15·24	15·55	15·85
15	13·50	13·84	14·17	14·51	14·85	15·18	15·52	15·86	16·19	16·53	16·87	17 21	17·54
15·5	14·89	15·27	15·64	16·01	16·38	16·76	17·13	17·50	17·87	18·25	18·62	18·99	19·36
16	16·38	16·79	17·20	17·61	18·02	18·43	18·84	19·25	19·66	20·07	20·48	20·89	21·30
16·5	17·97	18·42	18·87	19·31	19·76	20·21	20·66	21·11	21·56	22·01	22·46	22·91	23·36
17	19·65	20·14	20·63	21·12	21·62	22·11	22·60	23·09	23·58	24·07	24·56	25·06	25·55
17·5	21·44	21·97	22·51	23·04	23·58	24·12	24·65	25·19	25·72	26·26	26·80	27·33	27·87
18	23·33	23·91	24·49	25·08	25·66	26·24	26·83	27·41	27·99	28·58	29·16	29·74	30·33
18·5	25·33	25·96	26·59	27·22	27·86	28·49	29·12	29·76	30·39	31·02	31·66	32·29	32·92
19	27·44	28·12	28·81	29·49	30·18	30·86	31·55	32·24	32·92	33·61	34·29	34·98	35·67
19·5	29·66	30·40	31·14	31·88	32·62	33·37	34·11	34·85	35·59	36·33	37·07	37·81	38·56

There are several simpler methods of estimating approximately the wetted skin, such as :—

Mumford Rule, which is more accurate than Kirk's, is as follows :—

Rule 23.—Wetted skin $= (L \times D \times 1{\cdot}7) + (L \times B \times C)$

or $L(1{\cdot}7D + B.C)$.

L is the length between perpendiculars ; D is mean draught ; and B the beam, all in feet ; C is a factor which is the block co-efficient ; that is,

$$C = \frac{\text{displacement in cubic feet}}{L \times B \times D}$$

Seaton's Rules are,—

(*a*) **Rule 24.** Wetted skin $= (c \times d \times L) + \frac{D \times 35}{d}$

$c = 2$ area of immersed mid section $\div$ B $\times$ *d*.
For shallow draught ships $c = 2$.
For ships with high rise of floor $c = 1{\cdot}6$.
For ordinary ships whose draught of water is not less than $\frac{1}{4}$ the beam $c = 1{\cdot}8$.

D is the displacement in tons. *d* the mean moulded draught of water.

(*b*) **Rule 25.** Wetted skin $= 42\sqrt[4]{K} \times D^{\frac{3}{2}}$

$$42\sqrt[4]{K} = F\,;\quad K = L \div (0{\cdot}55\,B + d)$$

Where K is	4	the value of F is	59·4.
,,	5	,,	62·9.
,,	6	,,	65·7.
,,	8	,,	70·6.
,,	9	,,	72·8.
,,	10	,,	74·7.
,,	11	,,	76·4.

(*c*) **Rule 26.**—Suitable prism co-efficient for speed S,—

$$P_c = 0{\cdot}4\sqrt[4]{L} \div \sqrt[3]{S}.$$

(*d*) **Rule 27.**—Maximum economic speed for ship L feet long and prism co-efficient, F,—

$$S = (0{\cdot}4\sqrt[4]{L} \div P_c)^3.$$

(*e*) **Rule 28.**—Minimum length L for a speed S with a co-efficient prism F,—

$$L = \left(\frac{\sqrt[3]{S} \times P_c}{0{\cdot}4}\right)^4.$$

Speed and Power Curves, etc.

The most reliable method of determining the I.H.P. required to drive any proposed vessel at a given speed is to base the calculations upon the results obtained from the trials of "similar" vessels; the basis of which is the fact that "similar" vessels, driven at "corresponding" speeds, have the same co-efficient of performance, when the efficiency of the machinery is the same.

"Similar" vessels are those having the same ratio of length to breadth, and to draught, and the same degree of fineness; and "corresponding" speeds are those which are proportional to the square roots of the linear dimensions of the respective vessels (*e.g.* proportional to the square root of the lengths).

Froude found that the resistance of such vessels varied almost exactly as wetted surface × (speed)^2.

But to render the results of former trials readily accessible for such a purpose it is very desirable to have them plotted down as a series of curves, somewhat in the following manner :—

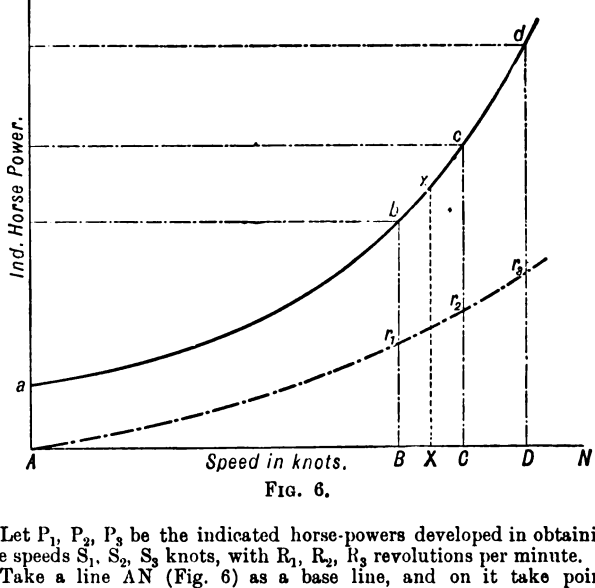


FIG. 6.

Let P_1, P_2, P_3 be the indicated horse-powers developed in obtaining the speeds S_1, S_2, S_3 knots, with R_1, R_2, R_3 revolutions per minute.

Take a line AN (Fig. 6) as a base line, and on it take points B, C, D, so that AB, AC, AD are proportional to S_1, S_2, S_3; at the points B, C, D erect ordinates B*b*, C*c*, D*d* proportional to P_1, P_2, P_3, and through the points *b*, *c*, and *d* draw the curve *d c b a*, which is called the "curve of power" or "curve of I.H.P." The nature of this curve is then such that if an ordinate be drawn through any other point X in the line AN, X*x* will represent the power necessary to obtain the speed AX from the same vessel, or another vessel of the same form and dimensions.

If the curve be accurately drawn, it will be found that it does not pass through A, but at a distance A*a* above that point, thus signifying that a certain amount of power is developed even at zero speed; A*a* thus represents the power required to overcome the constant friction of the engines. (See also page 17, under "Efficiency of Marine Machinery")

Similarly a curve of revolutions may be constructed by taking points r_1, r_2, r_3 in the ordinates so that B*r*, C*r*, D*r* are proportional to R_1, R_2, R_3.

The slip may also be shown by a curve whose ordinates are proportional to the slips at the speeds S_1, S_2, S_3.

Examination of the curves will show :—

(1) The I.H.P., revolutions, and slip corresponding to any speed intermediate to those observed;
(2) The constant friction, and therefore general efficiency of the engines;
(3) The suitability of the lines of the ship for the speeds,—a sudden rise of the curve towards the higher part, showing an undue increase of resistance at the higher speeds;
(4) The power of the speed with which the I.H.P. increases for the particular type of ship;
(5) The suitability of the propeller to the ship,—any sudden rise in the slip curve showing that the propeller is defective, either as regards diameter or surface, or both.

Perhaps the most useful curve, however, for the purpose of determining the power required to propel some "similar" ship, of different size, at any given speed, is one constructed as shown in fig. 7, where the abscissæ represent speeds in knots, and the ordinates numerical co-efficients of performance, derived either from the Admiralty formula $\left(\text{I.H.P.} = \frac{D^{\frac{2}{3}} \times S^3}{C}\right)$ or other similar expression.

Rule 28a.—Suppose fig. 7 to represent the curve given by a vessel 250 feet long and 2400 tons displacement, and that it is required to determine the power necessary to drive a "similar" vessel of 360 feet long and 7200 tons displacement at, say, 13 knots. The question then is,—what co-efficient of performance must be assumed?—and it is answered as follows:—

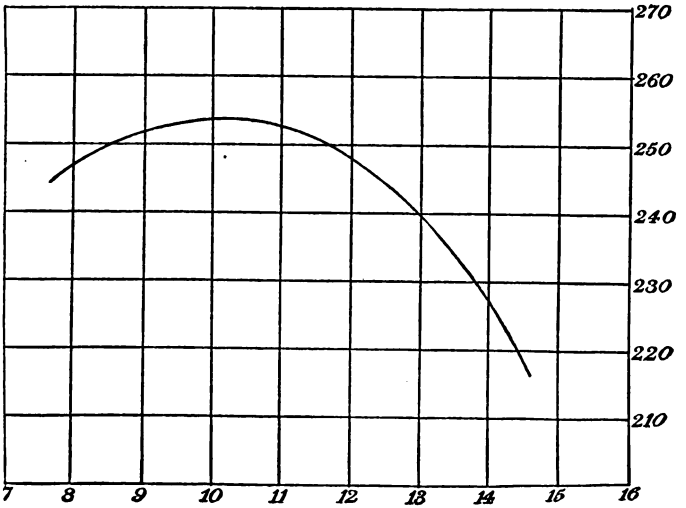


FIG. 7.

$$\sqrt{360} : \sqrt{250} : : 13 : x$$

$$\text{or } x = \frac{\sqrt{250} \times 13}{\sqrt{360}} = 10{\cdot}8 = \left\{ \begin{array}{l} \text{"corresponding" speed of} \\ \text{first, or type ship,} \end{array} \right.$$

and running up the ordinate for 10·8 knots until it cuts the curve, and then along the abscissa from this point of intersection the figure 253 is found, and this, when used in connection with the formula, gives 3250 as the required I.H.P.

In cases where there are no records of exactly "similar" ships, the value of the estimate made will of course depend very largely on the experience of the estimator.

Determination by Model Experiments.

Another method, employed by the Admiralty, and many of our leading firms of shipbuilders, for determining the power required to propel any new type of vessel is, to ascertain the resistance of a model of the new vessel in the experimental tank, and to calculate the power from the results obtained; and, where widely divergent types have to be dealt with, the method is no doubt of great value, but the expense is necessarily great.

The models are usually made of paraffin wax, from 12 feet to 20 feet long, and from ¾-inch to 1¼-inch thick; they are cast nearly to shape, and trimmed to the exact form on a shaping machine. The speeds and tension on the tow-rope are automatically recorded on paper drums driven by clock-work. The height and position of the waves created,—which are of special importance in the case of paddle vessels,—can also be noted and recorded.

The horse-power required is calculated from the resistance of the model, by the same principle of "corresponding" speeds referred to above, as follows:—

Let l and L = lengths of model and vessel, respectively;
v and V = corresponding speeds;
r and R = corresponding resistances;

$$\text{Then } \frac{V}{v} = \sqrt{\frac{L}{l}}$$

$$\text{and } \frac{R}{r} = \left(\frac{L}{l}\right)^3.$$

Example.—Suppose the E.H.P. nett or effective horse-power necessary to drive a ship 300 feet long at 15 knots is required. Let the length of the model be 12 feet, then the "corresponding" speed for it will be given by,—

$$v = 15 \sqrt{\frac{12}{300}} = 3 \text{ knots.}$$

Assume the resistance of the model at this speed to be 4 lbs., then resistance of ship at 15 knots will be,—

$$R = 4 \times \left(\frac{300}{12}\right)^3 = 62,500 \text{ lbs.};$$

$$\text{and } \frac{62,500 \times 15 \times 6080}{60 \times 33,000} = 2879;$$

or E.H.P. required is approximately 2879,—a slight correction having to be made for skin friction.

Co-efficients of Fineness.

The block co efficient expresses the ratio borne by the displacement volume to that of the parallelepiped circumscribing the immersed body.

Let V = displacement in cubic feet;
L = length on water line;
B = greatest immersed breadth;
D = draught of water of body (*ex* keel);
K = displacement co-efficient.

$$\text{Then } K = \frac{V}{L \times B \times D}.$$

The prismatic co-efficient,—which gives a truer measure of fineness of lines than the above,—expresses the ratio borne by the displacement volume to that of the prism swept by moving the immersed mid-ship section through the length at load water-line.

Generally speaking, the finer the water lines are, the easier is the ship driven at any speed, and consequently the speed co-efficients or multipliers vary inversely with the co-efficient of displacement. But as there is a limit to the speed at which a certain ship may be driven, any increase in power produces little or no increase in speed; there is a limit to the fineness for the lower speeds also, so that any decrease in the co-efficient of displacement causes little or no increase in the speed co-efficients—in other words, just as the form of a ship may be inefficient for high speeds, so it may be for low.

Table XIII. gives the prismatic co-efficients appropriate to all steamers from 100 ft. to 1000 ft. long, and from 10 knots to 28 knots speed, when of the usual ship form and style.

Table XIV. shows the values of the co-efficient C in the Admiralty speed formula $\text{I.H.P.} = \frac{D^{\frac{2}{3}} \times S^3}{C}$, when the ship is in agreement with Rules 26, 27, 28.

If a ship is finer than given by Rule 26, the value of C may be increased, as it may be also when the machinery has a higher efficiency than 92 per cent.

Table XV. gives the co-efficient for computing *effective* (*nett*) horse-power necessary to overcome skin friction based on Dr. Froude's constants.

Table XIII.—Prism Co-efficients of Ships suitable for Speeds and Lengths.

Length in feet.	Speed in Knots per Hour.																				
	10·0	10·5	11·0	11·5	12·0	12·5	13·0	14·0	15·0	16·0	17·0	18·0	19·0	20·0	21·0	22·0	23·0	24·0	25·0	26·0	27·0	28·0
100	·590	·578	·568	·560	·552	·545	·539	·525	·513	·501	·491	·483	...	...	...	...	...	...	...	...	...	...
125	·623	·611	·601	·592	·584	·576	·569	·554	·542	·531	·520	·509	·501	·492	·484	..	...	...	...	...	...	...
150	·651	·640	·629	·620	·611	·603	·595	·580	·567	·555	·544	·534	·525	·516	·507	·499	·492	·486	...	...	...	...
175	·677	·664	·654	·645	·636	·626	·618	·603	·589	·577	·566	·554	·545	·536	·528	·520	·512	·505	·500	·492	·485	...
200	·698	·687	·678	·667	·657	·648	·640	·624	·609	·596	·584	·574	·564	·553	·544	·535	·529	·522	·515	·508	·501	·495
225	·718	·706	·696	·687	·676	·666	·658	·643	·628	·615	·602	·590	·580	·571	·562	·553	·542	·534	·527	·520	·513	·506
250	·738	·725	·714	·702	·693	·685	·675	·659	·641	·630	·618	·607	·595	·585	·576	·568	·560	·552	·544	·537	·530	·523
275	·755	·742	·731	·720	·710	·700	·692	·674	·659	·646	·632	·620	·619	·599	·588	·579	·573	·563	·557	·550	·543	·536
300	·772	·760	·740	·737	·727	·716	·707	·690	·674	·660	·647	·634	·623	·613	·603	·594	·586	·578	·570	·563	·555	·548
325	·784	·773	·762	·752	·741	·731	·720	·703	·689	·672	·660	·648	·636	·626	·615	·606	·598	·590	·582	·574	·566	·559
350	·802	·790	·779	·769	·759	·748	·739	·720	·704	·684	·675	·663	·648	·640	·627	·618	·609	·601	·592	·584	·577	·570
375	·815	·803	·791	·780	·769	·759	·748	·730	·713	·697	·684	·671	·659	·650	·636	·627	·619	·611	·603	·595	·587	·580
400	·829	·816	·804	·792	·781	·771	·760	·743	·725	·709	·696	·681	·669	·658	·647	·638	·629	·621	·613	·604	·596	·590
450	·853	·840	·828	·816	·804	·793	·783	·764	·746	·731	·716	·702	·690	·678	·667	·656	·648	·640	·631	·622	·614	·601
500	·878	·863	·850	·837	·826	·815	·804	·784	·767	·750	·735	·721	·708	·697	·687	·677	·666	·657	·648	·640	·631	·620
550	·899	·885	·870	·858	·846	·834	·823	·805	·785	·768	·754	·739	·726	·713	·702	·693	·682	·672	·663	·654	·645	·637
600	·918	·904	·891	·877	·864	·853	·841	·821	·801	·785	·770	·756	·742	·729	·717	·707	·697	·687	·678	·670	·660	·650
650	·937	·923	·909	·896	·882	·871	·860	·838	·820	·801	·786	·771	·758	·743	·732	·720	·711	·701	·692	·682	·673	·664
700	·952	·940	·926	·910	·898	·886	·871	·855	·832	·816	·800	·785	·770	·759	·745	·734	·724	·714	·704	·695	·685	·677
750	...	·955	·942	·925	·912	·901	·890	·870	·847	·830	·814	·800	·783	·772	·758	·743	·736	·726	·716	·707	·697	·688
800	...	...	·959	·941	·929	·917	·902	·883	·861	·844	·828	·812	·800	·785	·771	·760	·750	·739	·729	·719	·709	·700
850	...	...	...	·956	·943	·931	·915	·900	·874	·857	·840	·824	·810	·797	·782	·771	·761	·750	·740	·729	·720	·710
900	...	...	...	...	·957	·945	·930	·910	·887	·870	·853	·837	·821	·809	·796	·783	·772	·761	·750	·740	·731	·721
950	...	...	...	...	...	·957	·940	·921	·900	·881	·864	·848	·832	·819	·805	·793	·782	·771	·760	·750	·741	·732
1,000	...	...	...	...	...	...	·954	·934	·912	·900	·876	·860	·843	·830	·815	·803	·792	·781	·770	·760	·751	·742

Table XIV.—Values of Co-efficient C in Formula I.H.P. $= \frac{D_S^{\frac{2}{3}} \times S^3}{C}$ when Co-efficient of Water Lines is not greater than 0·4 $\frac{\sqrt[4]{L}}{\sqrt[3]{S}}$.

Speed in knots.	10	12	14	16	18	20	21	22	23	24	25	26	27	28
100 feet long, . .	219	206	196	188	181	...	...	...	...	...	...	...	...	...
150 ,, . .	244	229	217	209	201	193	190	188	185	182	180	...	...	...
200 ,, . .	261	245	233	224	215	208	204	201	198	195	193	191	188	186
250 ,, . .	277	260	247	236	228	219	216	213	210	207	205	202	199	197
300 ,, . .	289	272	258	247	238	230	226	223	220	216	214	211	208	206
350 ,, . .	301	282	269	257	247	240	235	232	228	225	222	219	216	213
400 ,, . .	311	292	277	266	256	248	243	240	236	232	230	227	224	221
450 ,, . .	321	301	286	274	263	255	250	247	243	240	237	234	231	228
500 ,, . .	330	308	293	281	271	262	257	253	250	246	243	240	237	234
550 ,, . .	...	316	301	288	277	268	263	258	256	252	249	245	242	239
600 ,, . .	...	323	307	294	282	274	268	264	261	257	254	250	246	245
650 ,, . .	...	330	314	301	289	279	274	271	267	263	260	256	253	249
700 ,, . .	...	...	320	306	294	284	280	276	272	268	265	261	258	254
750 ,, . .	...	...	325	311	299	289	284	280	276	272	269	265	261	258
800 ,, . .	...	...	330	317	304	294	289	285	280	277	273	270	266	263
850 ,, . .	...	...	...	321	308	299	293	289	285	281	278	274	270	267
900 ,, . .	...	...	...	326	312	303	298	294	289	285	282	278	274	271

Immersed Surface Friction.

Table XVI. (page 47) gives a general statement of the results of Froude's experiments on this subject; they were made on boards $\frac{3}{4}$ in. thick and 19 in. deep, which were coated with the substances to be experimented on, and towed edgeways through the water. The resistances are given in lbs. per square foot at the standard speed of 600 feet per minute, and, as the power of the speed to which the friction is proportional is also given, the resistance at other speeds is easily calculated.*

Columns A give the power of the speed to which the resistance is approximately proportional; columns B give the mean resistance per square foot of the whole surface of a board of the lengths stated in the table; columns C give the resistance of a square foot of surface at the distance sternward from the cut-water stated in the heading.

Table XV.—Co-efficients for Computing Effective Horse-power required to overcome Skin Friction based on Mr Froude's Constants, as given by Mr A. W. Johns.

If S is the wetted surface in square feet, then

E.H.P. $= f$. S, where f has the values given below.

Speed in Knots.	Length of Ship in Feet.								
	100	150	200	250	300	350	400	450	500
25, .	·2516	·2477	·2458	·2444	·2434	·2415	·2415	·2407	·2399
24, .	·2242	·2207	·2190	·2178	·2169	·2160	·2152	·2145	·2138
23, .	·1988	·1957	·1942	·1931	·1923	·1916	·1908	·1902	·1895
22, .	·1753	·1726	·1713	·1703	·1696	·1690	·1683	·1677	·1672
21, .	·1537	·1514	·1502	·1494	·1487	·1481	·1476	·1471	·1466
20, .	·1340	·1319	·1308	·1301	·1296	·1291	·1286	·1281	·1277
19, .	·1159	·1141	·1132	·1126	·1121	·1171	·1112	·1108	·1105
18, .	·0995	·0979	·0972	·0966	·0962	·0958	·0955	·0951	·0948
17, .	·0846	·0833	·0827	·0822	·0819	·0815	·0812	·0810	·0807
16, .	·0713	·0702	·0697	·0693	·0690	·0687	·0685	·0682	·0680
15, .	·0594	·0585	·0580	·0577	·0575	·0573	·0570	·0568	·0567
14, .	·0489	·0481	·0478	·0475	·0473	·0471	·0469	·0468	·0466
13, .	·0397	·0390	·0387	·0385	·0384	·0382	·0381	·0379	·0378
12, .	·0315	·0312	·0309	·0308	·0307	·0305	·0304	·0303	·0302

In the above table skin friction is taken as varying as $V^{1\cdot825}$.

* NOTE.—See Rule 80.

Table XVI.—Resistances of Surfaces.

Nature of Surface.	Length of surface, or distance from cutwater, in feet.											
	2 Feet.			8 Feet.			20 Feet.			50 Feet.		
	A	B	C	A	B	C	A	B	C	A	B	C
Varnish, .	2·00	·41	·390	1·85	·325	·264	1·85	·278	·240	1·83	·250	·226
Paraffin, .	1·95	·38	·370	1·94	·314	·260	1·93	·271	·237	...	...	...
Tinfoil, .	2·16	·30	·295	1·99	·278	·263	1·90	·262	·244	1·83	·246	·232
Calice, . .	1·93	·87	·725	1·92	·626	·504	1·89	·531	·447	1·87	·474	·423
Fine sand,	2·00	·81	·690	2·00	·583	·450	2·00	·480	·384	2·06	·405	·337
Medium ,,	2·00	·90	·730	2·00	·625	·488	2·00	·534	·465	2·00	·488	·456
Coarse ,,	2·00	1·10	·880	2·00	·714	·520	2·00	·588	·490	..	...	...

True Mean Speed.

To determine the true mean speed of a vessel when the runs are taken on the measured mile, half with the tide, and half against :—

Example.

Runs.	Observed Speeds.	1st Means.	2nd Means.	3rd Means.	4th Means.	Mean of Means.
1st	18·5					
		15·90				
2nd	13·3		15·625			
		15·35		15·5750		
3rd	17·4		15·525		15·52500	
		15·70		15·4750		15·478125
4th	14·0		15·425		15·43125	True mean speed.
		15·15		15·3875		
5th	16·3		15·350			
		15·55				
6th	14·8		4)61·925			
	6)94·2		15·48125			
	15·70		Ordinary mean of second means.			
	Ordinary mean speed.					

The ordinary mean of second means is generally taken,—as unavoidable errors of observation render the third and following decimal places of very doubtful value.

Find the means of consecutive speeds continually found until only one remains.

If a, b, c, d, e, and f are the speeds as given of six runs on the measured mile, the ultimate mean of means may be found by the following rule.

Rule 29. Mean speed $= \frac{(a+f)+5\,(b+e)+10\,(c+d)}{32}$.

Taking the above example by this rule, the mean speed is 15·478 knots. It is, however, usual now when taking the speed as the ultimate mean of means to have an odd number of runs on the mile for greater accuracy in a tideway; then if five runs are made:—

$$\text{Mean Speed} = \frac{a+4b+6c+4d+e}{16}$$

Taking the first five runs in the above example, the mean speed is then 15·525.

Relation of Speeds and Powers.

Given two speeds of a vessel, and the corresponding horse-powers, to find what power of the speed the horse-power varies as:—

Let s and S = the two speeds.
" p and P = the corresponding powers.
" x = power of s and S that p and P vary as

Then $\frac{S^x}{s^x} = \frac{P}{p}$

or $x\,(\text{Log S} - \text{Log } s) = \text{Log P} - \text{Log } p$

Rule 30. And $x = \frac{\text{Log P} - \text{Log } p}{\text{Log S} - \text{Log } s}$.

Depth of Water for Speed Trials.

Dr D. W. Taylor's formula:

D is the draught of water of ship in feet.
L is her length in feet.
S the maximum speed in knots.

Rule 30a. Minimum depth $= \frac{10 \times D \times S}{L}$ fathoms.

Table XVII.—Speed in Knots from time of doing Nautical Mile.

Times and Speeds.

The number in this Table corresponding to the time in which a vessel passes over the measured mile is her speed in knots.

Secs.	2 min.	3 min.	4 min.	5 min.	6 min.	7 min.	8 min.	9 min.	10 min.	11 min.	12 min.	13 min.	14 min.
0	30·000	20·000	15·000	12·000	10·000	8·571	7·500	6·667	6·000	5·455	5·000	4·615	4·286
1	29·752	19·890	14·938	11·960	9·972	8·551	7·484	6·654	5·990	5·446	4·993	4·609	4·281
2	29·508	19·780	14·876	11·921	9·945	8·531	7·469	6·642	5·980	5·438	4·986	4·604	4·275
3	29·268	19·672	14·815	11·881	9·917	8·511	7·453	6·630	5·970	5·430	4·979	4·598	4·270
4	29·032	19·565	14·754	11·842	9·890	8·491	7·438	6·618	5·960	5·422	4·972	4·592	4·265
5	28·800	19·459	14·694	11·803	9·863	8·471	7·423	6·606	5·950	5·414	4·965	4·586	4·260
6	28·571	19·355	14·634	11·765	9·836	8·451	7·407	6·593	5·941	5·405	4·959	4·580	4·255
7	28·346	19·251	14·575	11·726	9·809	8·431	7·392	6·581	5·931	5·397	4·952	4·574	4·250
8	28·125	19·149	14·516	11·688	9·783	8·411	7·377	6·569	5·921	5·389	4·945	4·568	4·245
9	27·907	19·048	14·458	11·650	9·756	8·392	7·362	6·557	5·911	5·381	4·938	4·563	4·240
10	27·692	18·947	14·400	11·613	9·730	8·372	7·347	6·545	5·902	5·373	4·931	4·557	4·235
11	27·481	18·848	14·343	11·576	9·704	8·353	7·332	6·534	5·892	5·365	4·925	4·551	4·230
12	27·273	18·750	14·286	11·538	9·677	8·333	7·317	6·522	5·882	5·357	4·918	4·545	4·225
13	27·068	18·653	14·229	11·502	9·651	8·314	7·302	6·510	5·873	5·349	4·911	4·540	4·220
14	26·866	18·557	14·173	11·465	9·626	8·295	7·287	6·498	5·863	5·341	4·905	4·534	4·215
15	26·667	18·461	14·118	11·429	9·600	8·276	7·273	6·486	5·854	5·333	4·898	4·528	4·210
16	26·471	18·367	14·062	11·392	9·574	8·257	7·258	6·475	5·844	5·325	4·891	4·523	4·206
17	26·277	18·274	14·008	11·356	9·549	8·238	7·243	6·463	5·835	5·318	4·885	4·517	4·201
18	26·087	18·182	13·953	11·321	9·524	8·219	7·229	6·452	5·825	5·310	4·878	4·511	4·196
19	25·899	18·090	13·900	11·285	9·499	8·200	7·214	6·440	5·816	5·302	4·871	4·506	4·191
Secs.	2 min.	3 min.	4 min.	5 min.	6 min.	7 min.	8 min.	9 min.	10 min.	11 min.	12 min.	13 min.	14 min.

Should the speed be required in kilometres,—multiply the speed in knots, from the above Table, by 1·853.
See also Table CLXXX. of knots, miles, kilometres, &c.

Table XVII.—*continued.*

Times and Speeds.

The number in this Table corresponding to the time in which a vessel passes over the measured mile is her speed in knots.

Secs.	2 min.	3 min.	4 min.	5 min.	6 min.	7 min.	8 min.	9 min.	10 min.	11 min.	12 min.	13 min.	14 min.
20	25·714	18·000	13·846	11·250	9·474	8·182	7·200	6·429	5·806	5·294	4·865	4·500	4·186
21	25·582	17·910	13·793	11·215	9·449	8·163	7·186	6·417	5·797	5·286	4·858	4·494	4·181
22	25·352	17·822	13·740	11·180	9·424	8·145	7·171	6·406	5·788	5·279	4·852	4·489	4·176
23	25·175	17·734	13·688	11·146	9·399	8·126	7·157	6·394	5·778	5·271	4·845	4·483	4·171
24	25·000	17·647	13·636	11·111	9·375	8·108	7·143	6·383	5·769	5·263	4·839	4·478	4·167
25	24·828	17·561	13·585	11·077	9·351	8·090	7·129	6·372	5·760	5·255	4·832	4·472	4·162
26	24·658	17·476	13·534	11·043	9·326	8·072	7·115	6·360	5·751	5·248	4·826	4·466	4·157
27	24·490	17·391	13·483	11·009	9·302	8·054	7·101	6·349	5·742	5·240	4·819	4·461	4·152
28	24·324	17·308	13·433	10·976	9·278	8·036	7·087	6·338	5·732	5·233	4·813	4·455	4·147
29	24·161	17·225	13·383	10·942	9·254	8·018	7·073	6·327	5·723	5·225	4·806	4·450	4·143
30	24·000	17·143	13·333	10·909	9·231	8·000	7·059	6·316	5·714	5·217	4·800	4·444	4·138
31	23·841	17·062	13·284	10·876	9·207	7·982	7·045	6·305	5·705	5·210	4·794	4·439	4·133
32	23·684	16·981	13·235	10·843	9·184	7·965	7·031	6·294	5·696	5·202	4·787	4·433	4·128
33	23·529	16·901	13·187	10·811	9·160	7·947	7·018	6·283	5·687	5·195	4·781	4·428	4·124
34	23·377	16·822	13·139	10·778	9·137	7·930	7·004	6·272	5·678	5·187	4·774	4·423	4·119
35	23·226	16·744	13·091	10·746	9·114	7·912	6·990	6·261	5·669	5·180	4·768	4·417	4·114
36	23·077	16·667	13·043	10·714	9·091	7·895	6·977	6·250	5·660	5·172	4·762	4·412	4·110
37	22·930	16·590	12·996	10·682	9·068	7·877	6·963	6·239	5·651	5·165	4·756	4·406	4·105
38	22·785	16·514	12·950	10·651	9·045	7·860	6·950	6·228	5·643	5·158	4·749	4·401	4·100
39	22·642	16·438	12·903	10·619	9·023	7·843	6·936	6·218	5·634	5·150	4·743	4·396	4·096
Secs.	2 min.	3 min.	4 min.	5 min.	6 min.	7 min.	8 min.	9 min.	10 min.	11 min.	12 min.	13 min.	14 min.

Should the speed be required in kilometres,—multiply the speed in knots, from the above Table, by 1·853.
See also Table CLXXX. of knots, miles, kilometres, &c.

Times and Speeds.

The number in this Table corresponding to the time in which a vessel passes over the measured mile is her speed in knots.

Secs.	1 min.	2 min.	3 min.	4 min.	5 min.	6 min.	7 min.	8 min.	9 min.	10 min.	11 min.	12 min.	13 min.	14 min.
40	36·000	22·500	16·364	12·857	10·588	9·000	7·826	6·923	6·207	5·625	5·143	4·737	4·390	4·091
41	35·644	22·360	16·290	12·811	10·557	8·978	7·809	6·910	6·196	5·616	5·136	4·731	4·385	4·086
42	35·294	22·222	16·216	12·766	10·526	8·955	7·792	6·897	6·186	5·607	5·128	4·724	4·379	4·082
43	34·951	22·086	16·143	12·721	10·496	8·933	7·775	6·883	6·175	5·599	5·121	4·718	4·374	4·077
44	34·615	21·951	16·071	12·676	10·465	8·911	7·759	6·870	6·164	5·590	5·114	4·712	4·369	4·072
45	34·286	21·818	16·000	12·632	10·435	8·889	7·742	6·857	6·154	5·581	5·106	4·706	4·364	4·068
46	33·962	21·687	15·929	12·587	10·405	8·867	7·725	6·844	6·143	5·573	5·099	4·700	4·358	4·063
47	33·644	21·557	15·859	12·544	10·375	8·845	7·709	6·831	6·133	5·564	5·092	4·693	4·353	4·059
48	33·333	21·429	15·789	12·500	10·345	8·824	7·692	6·818	6·122	5·556	5·085	4·687	4·348	4·054
49	33·028	21·302	15·721	12·457	10·315	8·802	7·676	6·805	6·112	5·547	5·078	4·681	4·343	4·049
50	32·727	21·176	15·652	12·414	10·286	8·780	7·660	6·792	6·102	5·538	5·070	4·675	4·337	4·045
51	32·432	21·053	15·584	12·371	10·256	8·759	7·643	6·780	6·091	5·530	5·063	4·669	4·332	4·040
52	32·143	20·930	15·517	12·329	10·227	8·738	7·627	6·767	6·081	5·521	5·056	4·663	4·327	4·035
53	31·858	20·809	15·451	12·287	10·198	8·717	7·611	6·754	6·071	5·513	5·049	4·657	4·322	4·031
54	31·579	20·690	15·385	12·245	10·169	8·696	7·595	6·742	6·061	5·505	5·042	4·651	4·316	4·027
55	31·304	20·571	15·319	12·203	10·141	8·675	7·579	6·729	6·050	5·496	5·035	4·645	4·311	4·022
56	31·034	20·455	15·254	12·162	10·112	8·654	7·563	6·716	6·040	5·488	5·028	4·639	4·306	4·018
57	30·769	20·339	15·190	12·121	10·084	8·633	7·547	6·704	6·030	5·479	5·021	4·633	4·301	4·013
58	30·508	20·225	15·126	12·081	10·056	8·612	7·531	6·691	6·020	5·471	5·014	4·627	4·296	4·009
59	30·252	20·112	15·063	12·040	10·028	8·592	7·516	6·679	6·010	5·463	5·007	4·621	4·291	4·004
Secs.	1 min.	2 min.	3 min.	4 min.	5 min.	6 min.	7 min.	8 min.	9 min.	10 min.	11 min.	12 min.	13 min.	14 min.

Should the speed be required in kilometres,—multiply the speed in knots, from the above Table, by 1·853.
See also Table CLXXX. of knots, miles, kilometres, &c.

The following Table of the ⅔ powers of numbers will be of service in all calculations for which displacement is taken as a basis.

Table XVIII.—Two-thirds powers of numbers.

Number.	⅔rd power.	Number.	⅔rd power.	Number.	⅔rd power.	Number.	⅔rd power.
100	21·54	480	61·30	860	90·43	1240	115·42
110	22·96	490	62·15	870	91·13	50	116·04
120	24·33	500	62·99	880	91·83	60	116·66
130	25·66	510	63·83	890	92·52	70	117·27
140	26·96	520	64·66	900	93·22	80	117·89
150	28·23	530	65·49	910	93·91	90	118·50
160	29·47	540	66·31	920	94·59	1300	119·11
170	30·69	550	67·13	930	95·28	10	119·72
180	31·88	560	67·94	940	95·96	20	120·33
190	33·05	570	68·74	950	96·64	30	120·94
200	34·21	580	69·54	960	97·32	40	121·55
210	35·33	590	70·34	970	97·99	50	122·15
220	36·44	600	71·13	980	98·66	60	122·75
230	37·54	610	71·92	990	99·33	70	123·35
240	38·62	620	72·71	1000	100·00	80	123·95
250	39·68	630	73·49	10	100·66	90	124·55
260	40·74	640	74·26	20	101·33	1400	125·14
270	41·78	650	75·03	30	101·99	10	125·74
280	42·80	660	75·80	40	102·65	20	126·33
290	43·81	670	76·57	50	103·30	30	126·92
300	44·81	680	77·33	60	103·96	40	127·51
310	45·80	690	78·08	70	104·61	50	128·10
320	46·78	700	78·84	80	105·26	60	128·69
330	47·75	710	79·59	90	105·91	70	129·28
340	48·71	720	80·33	1100	106·56	80	129·87
350	49·66	730	81·07	10	107·20	90	130·45
360	50·61	740	81·81	20	107·85	1500	131·03
370	51·54	750	82·55	30	108·49	10	131·61
380	52·46	760	83·28	40	109·13	20	132·19
390	53·38	770	84·01	50	109·76	30	132·77
400	54·29	780	84·73	60	110·40	40	133·35
410	55·19	790	85·46	70	111·03	50	133·93
420	56·08	800	86·18	80	111·67	60	134·50
430	56·97	810	86·89	90	112·30	70	135·08
440	57·85	820	87·61	1200	112·92	80	135·65
450	58·72	830	88·32	10	113·55	90	136·23
460	59·59	840	89·03	20	114·17	1600	136·80
470	60·45	850	89·73	30	114·80	10	137·37

Table XVIII.—Two-thirds powers of numbers—*continued.*

Number.	⅔rd power.	Number.	⅔rd power.	Number.	⅔rd power.	Number.	⅔rd power.
1620	137·93	2080	162·94	2920	204·28	3760	241·80
30	138·50	2100	163·99	40	205·22	80	242·65
40	139·06	20	165·02	60	206·15	3800	243·51
50	139·63	40	166·05	80	207·08	20	244·36
60	140·19	60	167·09	3000	208·01	40	245·22
70	140·75	80	168·12	20	208·93	60	246·07
80	141·32	2200	169·15	40	209·85	80	246·91
90	141·88	20	170·17	60	210·76	3900	247·76
1700	142·44	40	171·19	80	211·68	20	248·61
10	143·00	60	172·20	3100	212·59	40	249·45
20	143·55	80	173·22	20	213·51	60	250·29
30	144·11	2300	174·24	40	214·42	80	251·14
40	144·66	20	175·24	60	215·33	4000	251·98
50	145·22	40	176·25	80	216·24	20	252·82
60	145·77	60	177·25	3200	217·15	40	253·65
70	146·32	80	178·26	20	218·05	60	254·49
80	146·87	2400	179·26	40	218·95	80	255·33
90	147·42	20	180·25	60	219·85	4100	256·16
1800	147·97	40	181·24	80	220·75	20	257·00
10	148·52	60	182·23	3300	221·65	40	257·83
20	149·06	80	183·22	20	222·54	60	258·67
30	149·61	2500	184·20	40	223·44	80	259·49
40	150·15	20	185·18	60	224·34	4200	260·31
50	150·70	40	186·16	80	225·22	20	261·14
60	151·24	60	187·14	3400	226·11	40	261·96
70	151·78	80	188·11	20	226·99	60	262·78
80	152·32	2600	189·08	40	227·88	80	263·60
90	152·86	20	190·05	60	228·76	4300	264·42
1900	153·40	40	191·02	80	229·64	20	265·24
10	153·94	60	191·98	3500	230·52	40	266·06
20	154·47	80	192·93	20	231·40	60	266·87
30	155·01	2700	193·89	40	232·27	80	267·69
40	155·54	20	194·85	60	233·14	4400	268·51
50	156·08	40	195·80	80	234·02	20	269·32
60	156·61	60	196·75	3600	234·89	40	270·13
70	157·14	80	197·71	20	235·76	60	270·95
80	157·68	2800	198·66	40	236·62	80	271·76
90	158·21	20	199·60	60	237·49	4500	272·56
2000	158·74	40	200·54	80	238·36	20	273·37
20	159·79	60	201·48	3700	239·22	40	274·17
40	160·84	80	202·42	20	240·08	60	274·98
60	161·89	2900	203·35	40	240·94	80	275·78

Table XVIII.—Two-thirds powers of numbers—*continued*.

Number.	⅔rd power.	Number.	⅔rd power.	Number.	⅔rd power.	Number.	⅔rd power.
4600	276·58	6150	335·67	8300	409·93	10,900	491·61
20	277·39	6200	337·49	50	411·57	11,000	494·61
40	278·19	50	339·30	8400	413·22	100	497·60
60	278·99	6300	341·11	50	414·85	200	500·58
80	279·78	50	342·91	8500	416·49	300	503·56
4700	280·58	6400	344·71	50	418·12	400	506·53
20	281·38	50	346·50	8600	419·75	500	509·48
40	282·17	6500	348·29	50	421·37	600	512·43
60	282·96	50	350·07	8700	423·00	700	515·38
80	283·76	6600	351·85	50	424·62	800	518·31
4800	284·55	50	353·62	8800	426·24	900	521·23
20	285·33	6700	355·39	50	427·85	12,000	524·15
40	286·11	50	357·16	8900	429·46	100	527·05
60	286·90	6800	358·93	50	431·06	200	529·95
80	287·68	50	360·68	9000	432·67	300	532·83
4900	288·47	6900	362·43	50	434·27	400	535·72
20	289·26	50	364·18	9100	435·86	500	538·60
40	290·05	7000	365·93	50	437·45	600	541·48
60	290·84	50	367·67	9200	439·04	700	544·34
80	291·62	7100	369·41	50	440·64	800	547·20
5000	292·40	50	371·13	9300	442·23	900	550·04
50	294·34	7200	372·86	50	443·82	13,000	552·88
5100	296·27	50	374·58	9400	445·40	100	555·70
50	298·21	7300	376·31	50	446·97	200	558·53
5200	300·15	50	378·02	9500	448·54	300	561·35
50	302·06	7400	379·74	50	450·11	400	564·16
5300	303·98	50	381·44	9600	451·68	500	566·96
50	305·89	7500	383·15	50	453·25	600	569·76
5400	307·80	50	384·85	9700	454·82	700	572·54
50	309·68	7600	386·55	50	456·39	800	575·33
5500	311·58	50	388·24	9800	457·95	900	578·10
50	313·46	7700	389·93	50	459·50	14,000	580·88
5600	315·34	50	391·62	9900	461·06	100	583·63
50	317·21	7800	393·30	50	462·61	200	586·38
5700	319·09	50	394·98	10,000	464·16	300	589·13
50	320·95	7900	396·66	100	467·25	400	591·88
5800	322·81	50	398·33	200	470·33	500	594·61
50	324·66	8000	400·00	300	473·39	600	597·34
5900	326·51	50	401·66	400	476·44	700	600·07
50	328·35	8100	403·32	500	479·49	800	602·80
6000	330·19	50	404·97	600	482·54	900	605·51
50	332·02	8200	406·63	700	485·57	15,000	608·22
6100	333·85	50	408·28	800	488·60	15,100	610·92

Table XVIII.—Two-thirds powers of numbers—*continued.*

Number.	⅔rd power.	Number.	⅔rd power.	Number.	⅔rd power.	Number.	⅔rd power.
15,200	613·61	19,500	724·5	23,800	827·4	28,100	924·3
300	616·30	600	726·9	900	829·7	200	926·5
400	618·98	700	729·4	24,000	832·0	300	928·7
500	621·66	800	731·9	100	834·4	400	930·9
600	624·33	900	734·4	200	836·7	500	933·0
700	627·00	20,000	736·8	300	839·0	600	935·2
800	629·66	100	739·2	400	841·3	700	937·4
900	632·32	200	741·7	500	843·6	800	939·6
16,000	634·97	300	744·2	600	845·9	900	941·7
100	637·61	400	746·6	700	848·1	29,000	943·9
200	640·24	500	749·0	800	850·4	100	946·1
300	642·87	600	751·5	900	852·7	200	948·3
400	645·50	700	753·9	25,000	855·0	300	950·4
500	648·12	800	756·3	100	857·3	400	952·6
600	650·74	900	758·8	200	859·5	500	954·7
700	653·35	21,000	761·2	300	861·8	600	956·9
800	655·96	100	763·0	400	864·1	700	959·1
900	658·56	200	766·0	500	866 4	800	961·2
17,000	661·15	300	768·4	600	868·6	900	963·4
100	663·74	400	770·8	700	870·9	30,000	965
200	666·33	500	773·2	800	873·1	250	970
300	668·91	600	775·6	900	875·4	500	976
400	671·48	700	778·0	26,000	877·6	750	981
500	674·05	800	780·6	100	879·9	31,000	987
600	676·62	900	782·8	200	882·1	250	992
700	679·18	22,000	785·1	300	884·4	500	997
800	681·74	100	787·5	400	886·6	750	1002
900	684·29	200	789·9	500	888·9	32,000	1008
18,000	686·83	300	792·3	600	891·1	250	1013
100	689·37	400	794·6	700	893·3	500	1018
200	691·91	500	797·0	800	895·6	750	1023
300	694·44	600	799·4	900	897·8	33,000	1029
400	696·97	700	801·7	27,000	900·0	250	1034
500	699·49	800	804·1	100	902·2	500	1039
600	702·01	900	806·4	200	904·4	750	1045
700	704·52	23,000	808·8	300	906·7	34,000	1050
800	707·03	100	811·1	400	908·8	250	1055
900	709·54	200	813·4	500	911·1	500	1060
19,000	712·0	300	815·8	600	913·3	750	1065
100	714·5	400	818·1	700	915·5	35,000	1070
200	717·0	500	820·4	800	917·7	500	1080
300	719·5	600	822·8	900	919·9	36,000	1090
400	722·0	700	825·1	28,000	922·1	500	1100

Table XVIII.—Two-thirds powers of numbers—*continued.*

Number.	⅔rd power.	Number.	⅔rd power.	Number.	⅔rd power.	Number.	⅔rd power.
37,000	1110	40,500	1179	44,000	1246	50,000	1357
500	1120	41,000	1189	500	1256	51,000	1375
38,000	1130	500	1199	45,000	1265	52,000	1393
500	1140	42,000	1208	46,000	1284	53,000	1411
39,000	1150	500	1218	47,000	1302	54,000	1428
500	1160	43,000	1227	48,000	1321	55·000	1446
40,000	1170	500	1237	49,000	1339	56·000	1463

TRIPLE VERSUS COMPOUND ENGINES.

The great economy resulting from the use of high-pressure steam is due to the fact that the increased pressure is obtained by the expenditure of an amount of heat quite insignificant compared with the additional amount of energy it renders available. The following tabular statement will make this clear at a glance :—

Table XIX.—Steam Pressures and Efficiencies.

Working pressures lbs. per sq. in.	Absolute pressure lbs. per sq. in.	Heat added commences from	Evaporation effected at	Total heat of evaporation in Thermal units per pound.	Specific volume cubic feet per pound.	Total effective external work by one pound of steam in ft.-lbs.	Heat used per 1000 ft.-lbs. of work done. Thermal units.	Theoretical saving of fuel at each step per cent.
30	45	100° F.	274° F.	1098	9·26	126,256	8·696	...
75	90	100° F.	320° F.	1112	4·81	174,472	6·373	26·7
125	140	100° F.	353° F.	1122	3·17	206,994	5·420	14·9
175	190	100° F.	377° F.	1129	2·38	231,200	4·883	10·8
225	240	100° F.	397° F.	1135	1·912	255,000	4·425	8·4

Note.—The release pressure is taken to be 10 lbs. per square inch, absolute, in each case, and a back pressure of 4 lbs. per square inch, absolute, is assumed.

That is to say,—an engine using steam of 125 lbs. pressure should, other things being equal, consume nearly 15 per cent. less fuel than one using steam of 75 lbs. pressure; and one using steam of 175 lbs. pressure should effect a further saving of nearly 11 per cent.,—so that, by using 175 lbs. steam in place of 75 lbs., a saving of over 23 per cent. should be effected.

In practice, however, a two-stage compound engine, using steam of 140 lbs. pressure, shows very little economy over a simple engine using 90 lbs. steam; whilst a triple-stage engine using 140 lbs. steam shows even a greater economy than is theoretically due to the increased pressure. This is due chiefly to the diminished range of temperature in each cylinder, and the low pressure on pins, guides, and bearings.

The actual amount of fuel saved by using a triple engine, working at 160 lbs. pressure, in place of a compound engine using, say, 75 lbs. steam, is nearly 25 per cent.; and a quadruple engine with 200 lbs. saves more still, due to the higher pressure and greater number of stages of expansion.

When the average magnitudes of the stresses set up in the two types of engine are compared, the result is again greatly in favour of the triple.

Compare a three-crank triple engine with an ordinary compound, having same size of L.P. cylinder, same length of stroke, and developing the same power,—the one using steam at 150 lbs., and the other at 75 lbs.; also let the referred mean pressure be 24 lbs. in each case, and the efficiency of the expansion be the same in the two cases. Then, further, let the L.P. piston area be represented in each case by the number 14; the H.P. area of triple by 2, and the H.P. area of compound by 4, and the M.P. area of triple by 5. Then, if the equivalent, or referred, mean pressure is equally obtained from the cylinders in each case, the relative work done will be as follows:—

Triple Engine.		Compound Engine.	
H.P. cylinder,	$2 \times \left(\frac{24}{3} \times \frac{14}{2}\right)$ or 112.	H.P. cylinder,	$4 \times \left(\frac{24}{2} \times \frac{14}{4}\right)$ or 168.
M.P. ,,	$5 \times \left(\frac{24}{3} \times \frac{14}{5}\right)$ or 112.	L.P. ,,	$14 \times \frac{24}{2}$ or 168.
L.P. ,,	$14 \times \frac{24}{3}$ or 112.		

That is,—the average stress on the rods, columns, guides, &c., is 50 per cent. more with the compound than with the triple engine. The triple engine is also better balanced than the compound, works with less vibration, gives a more equable turning moment, and consequently a higher efficiency of propeller.

It is not surprising, therefore, that the triple compares so favourably with the compound as regards wear and tear.

RATIOS OF CYLINDERS.

Two-Stage or Compound Engines.—Engines of this type are still commonly fitted in paddle vessels, and in small screw steamers; as also for many auxiliary purposes, such as for driving independent air and circulating pumps, feed pumps, dynamos, &c.

When fitted in **Paddle vessels**, the boiler pressure is usually from 70 to 100 lbs. for oscillating engines, and from 90 to 120 lbs. for other types,—the latter pressure mostly with diagonal engines.

But considerations of weight and bulk preclude the adoption of high ratios, and experience has shown that a ratio of about 1 : 3·25 for

pressures up to 110 lbs., and for 140 lbs. 1 : 3·8 is quite satisfactory. The extra quantity of coal to be carried is of little consequence where the voyage is only three or four hours' duration.

The ratios of cylinders for **auxiliary engines** are also determined more from considerations of weight and bulk than of economy, and with little regard to boiler pressures,—with the result that they are rarely made so great as in the case of the propelling engines, even when the auxiliary engine must run hour for hour with the main engines; in practice this ratio is generally about 1 : 2·5 to 1 : 3 for cylinders side by side, and 1 : 3·5 to 1 : 4 for tandem engines. In Naval practice, when the boiler pressure used was 300 lbs., the cylinders of compound auxiliary engines were only made 4 to 1.

Three-stage or **Triple engines**, and **Four-Stage** or **Quadruple engines.**—In the case of main or propelling engines for merchant or mail steamers, space and weight are, within ordinary limits, of very little consequence, whilst coal consumption is of the utmost importance; L.P. cylinders are therefore usually large enough to expand the steam to the full economical limit,—of say, 6 lbs. absolute.

The ratios of cylinders depend largely on the rate of expansion at full speed. The following holds good in practice:—

$$\text{Rate of expansion} = \frac{\text{absolute pressure}}{Q}.$$

For Naval engines, $Q = 21$.
In express steamers short voyage, $Q = 19$.
,, ,, long ,, $Q = 18$.
Passenger cargo steamers, $Q = 17$.
Tramp cargo steamers, $Q = 16$.

The following Table shows the ratios of cylinders necessary to effect this, with a cut-off in the H.P. of ·6 of the stroke:—

Table XX.—Ratios of Cylinders in multiple Stage Reciprocators.

Absolute pressure.	125	135	145	155	165	175	185	195	205	215	225
Ratio $\frac{\text{M.P.}}{\text{H.P.}}$	2·04	2·11	2·19	2·26	2·33	2·40	2·47	2·54	2·61	2·68	2·75
Ratio $\frac{\text{L.P.}}{\text{H.P.}}$	4·4	4·75	5·1	5·43	5·8	6·13	6·5	6·8	7·18	7·53	7·9
Working pressure (above atmosphere)	110	120	130	140	150	160	170	180	190	200	210

The values given in the above table for the ratio $\frac{\text{L.P.}}{\text{H.P.}}$ are such that the nominal rate of expansion is, in each case, $\frac{\text{absolute pressure}}{17}$; and those for ratio $\frac{\text{M.P.}}{\text{L.P.}}$ are calculated by the formula,—

Rule 31. $$\text{M.P. area} = \frac{\text{L.P. area}}{1{\cdot}1 \times \sqrt{\text{Ratio of H.P. to L.P.}}}$$

When the working pressure exceeds 195 lbs. absolute for economy in consumption, the steam should be expanded in four stages, the engine should be a Quadruple.

For Quadruple engines, working at 200 lbs. to 215 lbs. pressure, the relative areas of cylinders may be about 1 : 1·8 : 3·6 : 7·9 ;

$$\text{or } \frac{\text{1st M.P.}}{\text{H.P.}} = 1{\cdot}8,$$

$$\frac{\text{2nd M.P.}}{\text{1st M.P.}} = 2,$$

$$\text{and } \frac{\text{L.P.}}{\text{2nd M.P.}} = 2{\cdot}2.$$

It is possible to arrange the cylinders of large engines in many different ways, but the designer must always keep in view equality of stresses, ranges of temperature in the various cylinders, and also the best disposition of the reciprocating weights for reducing vibration.

Figs. 8, 8A, 8B, and 8C show some of these arrangements.

If possible, three cranks should be employed in preference to two, and the tandem arrangement only be resorted to in cases where there would otherwise be a low-pressure cylinder of very large size,—say over 100 inches diameter, and where four cranks are inadmissible.

Naval Engines.—In war-vessels economy at full power is only a secondary consideration, whilst weight of machinery and space occupied by it are of the first importance, and the ratio of L.P. piston to H.P. is therefore seldom so great as in the merchant service. While the boiler pressure used in the Navy stood at 155 lbs., the piston areas were usually made of the proportions 1 : 2 to 2·25 : 4·7 to 5. With a working pressure of 250 lbs. at the engines the proportions used have been about 1 : 2·65 : 7. In passenger steamers and war-vessels with relatively high revolution speeds, the necessity of balancing the inertia forces to reduce vibration to a minimum has led to the use of four-crank triple and four-crank quadruple engines.

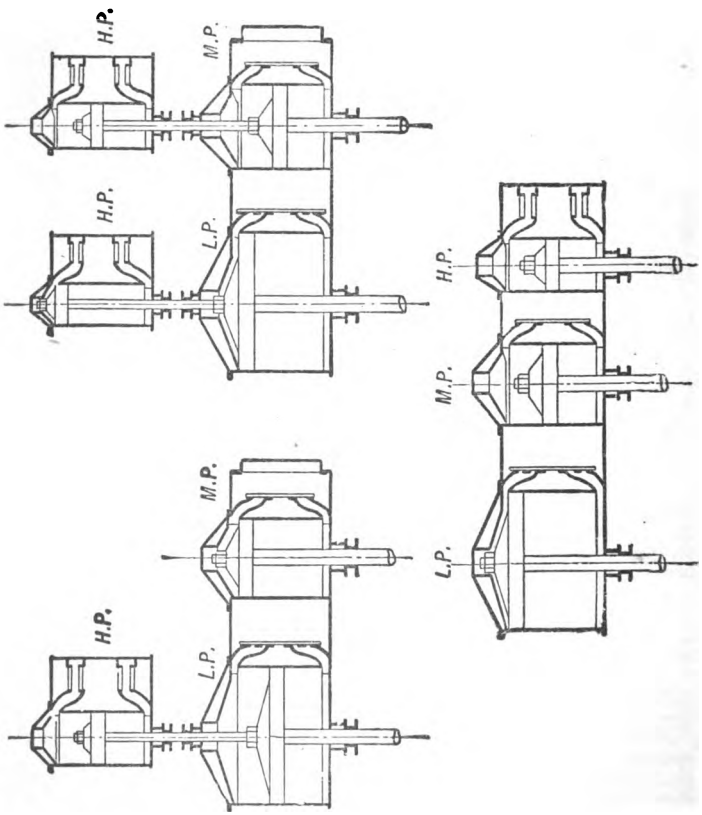


FIG. 8 Triple Expansion Engines.

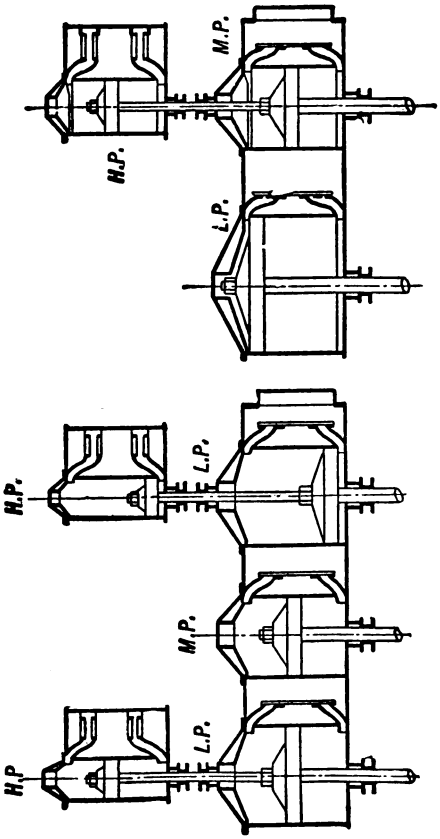


FIG. 8A. Triple Expansion Engines.

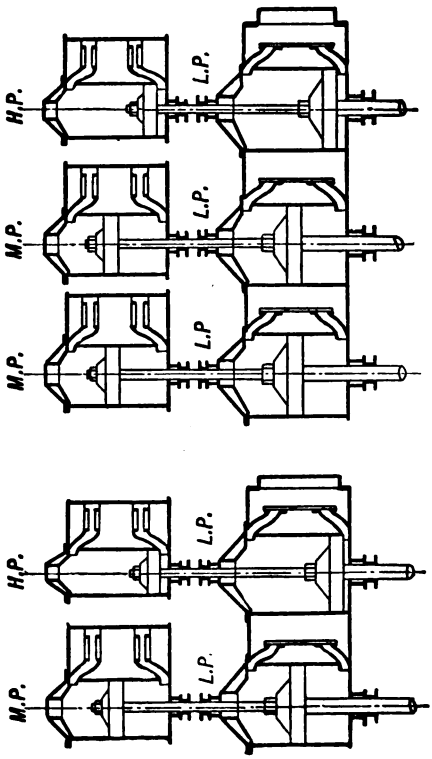


FIG. 68. Triple Expansion Engines.

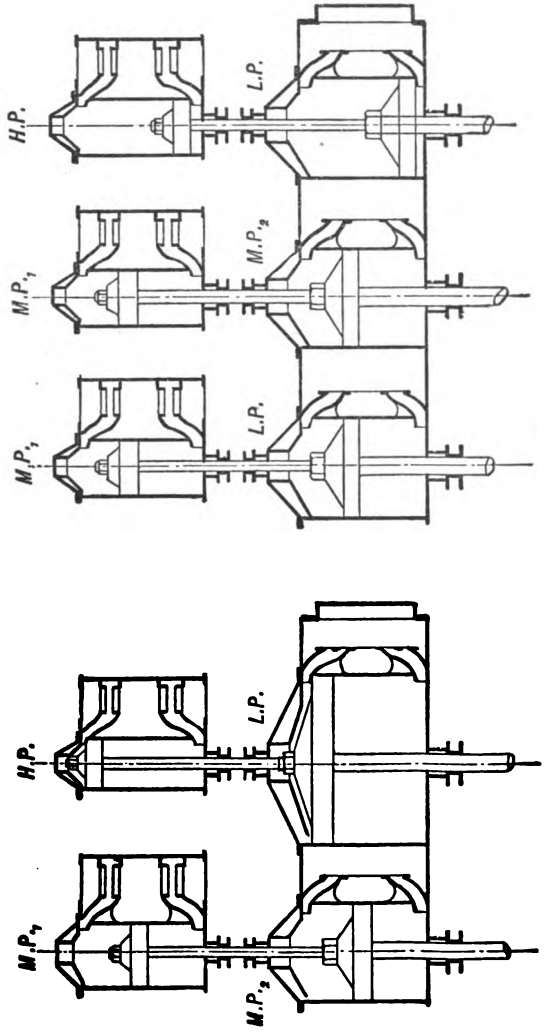


FIG. 8C. Quadruple Expansion Engines.

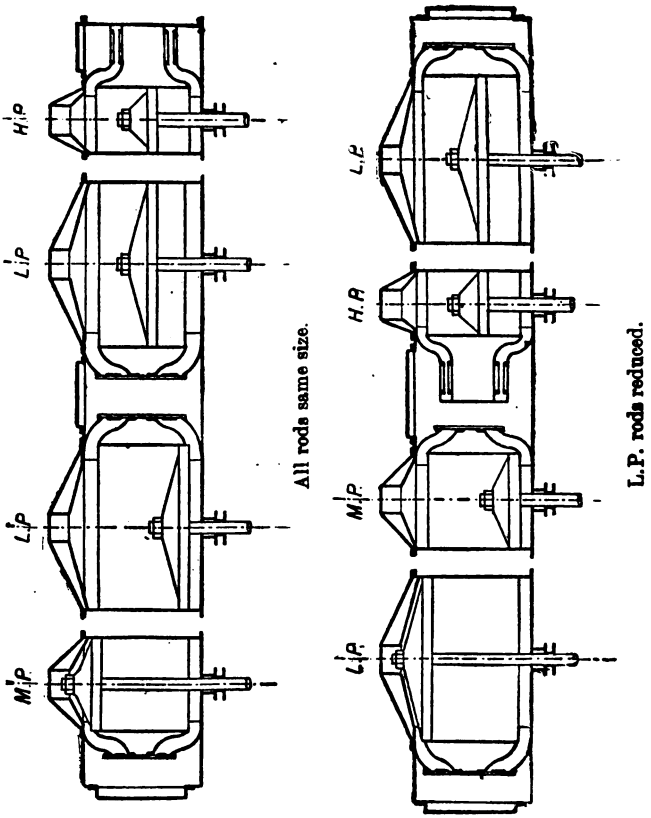


FIG. 8D.—Four-crank Triple Expansion Engines.

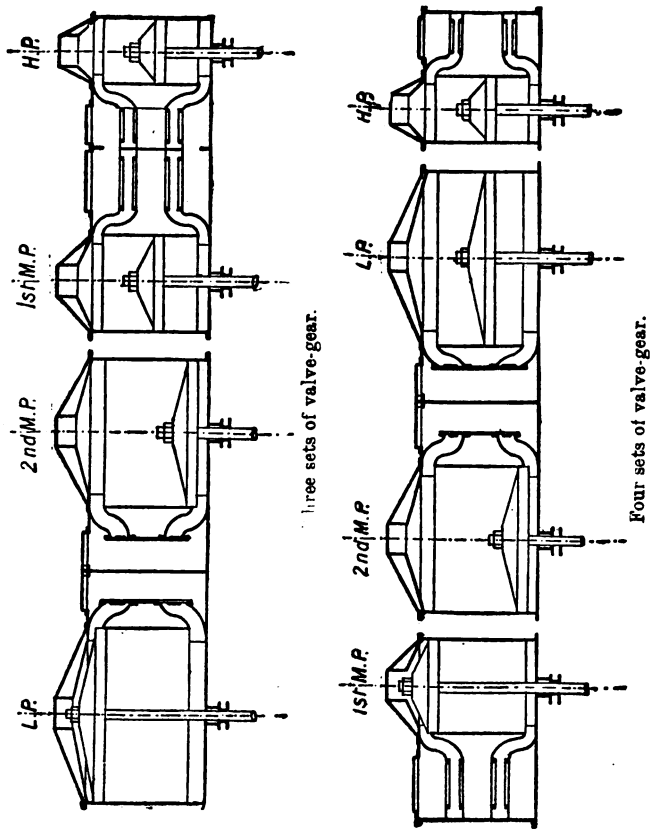


FIG. 8E.—Four-crank Quadruple Expansion Engines.

EQUIVALENT MEAN PRESSURES, ETC.

In estimating the mean pressure in any multiple-stage engine, it is usual *to refer it to the L.P. cylinder, i.e.* to calculate the pressure that would be required if the work were all done in the L.P. cylinder only.

The value of this equivalent—or, as it is called, *referred*—mean pressure for a triple engine is therefore,—

Rule 32.

$$Rp_m = \text{L.P. mean press.} + \frac{\text{M.P. mean press.}}{\text{Ratio of L.P. to M.P.}} + \frac{\text{H.P. mean press.}}{\text{Ratio of L.P. to H.P.}}$$

Records of the Rp_m obtained with each engine should be kept,—together with notes of the various conditions under which the trials were made,—such as pressure of steam at H.P. valve casing, degree of opening of throttle or regulator valve, amount gear is "linked-up," vacuum in condenser, &c.,—as they supply a ready means of checking calculations for sizes of cylinders, cuts off, &c., of proposed engines of the same type, and of determining the exact values of the factors, approximate values of which are given in Table XXIV. page 73.

Expansion of Steam: to determine Mean Pressures.

The mean pressure obtained, and consequently the work done, in any steam cylinder, or series of steam cylinders, depends mainly on the absolute pressure and the condition of the steam at admission, also the rate of expansion at release.

The general formula for pressure and volume of steam expanding under conditions obtaining in practice is

$$pv^{\epsilon} = \text{constant.}$$

When the cylinders are jacketed so as to render heat to the steam rather than abstract it during expansion,—

(i.) Steam as supplied from the ordinary boiler without superheater or other means of drying, $\epsilon = 1{\cdot}00$.

(ii.) When the steam is supplied to such cylinders "dry" and there is no condensation during expansion, $\epsilon = 1{\cdot}0625$ or $^{17}/_{16}$ths.

(iii.) When the steam to these cylinders is superheated and the temperature is kept constant during expansion (*isothermic*), then $\epsilon = 1{\cdot}00$.

When the cylinders are not jacketed but well clothed, as in general marine practice,—

(iv.) Steam supplied simply as produced saturated $\epsilon = 1{\cdot}111$, ($^{10}/_{9}$ths).

(v.) If superheated steam supplied $\epsilon = 1{\cdot}180$.

Mean Pressure of Steam Expanding Isothermally, as in a Steam Cylinder when Jacketed.

Let V be the volume, P the pressure, and T the temperature of steam as supplied.

,, v be the volume, p the pressure, t the temperature at the end of expansion.

,, v_0 be the volume, p_0 the pressure, and t_0 the temperature of back pressure.

R is the ratio of expansion $=\frac{P}{p}$; and PV = constant.

A. (1) Mean pressure due to a pressure P, and a rate of expansion R.

Rule 33. $P_m=\frac{1+\log_e R}{R}\times P-p_0$ ($p_0=o$, *vide* Table XXI.)

If the back pressure in a modern condenser be taken as 1 lb., then for a full expansion, as in the case of a turbine, R = P.

(2) $P_m=\frac{1+\log_e R}{R}\times R-1$; or $P_m=\log_e R$ or $\log_e P$.

(3) The work done by a pound of steam during admission and expansion from pressure P to 1 lb. will be found as follows:—

Maximum work of 1 lb. of steam of pressure $P=\log_e P\times 144\times v$; in this case $v=332$. Then

$$\text{maximum work}=47{,}808\log_e P.$$

(4) If steam expands to a pressure p and then drops to p_0, which is the usual condition of working in a reciprocating engine,

$$\text{Mean pressure}=\frac{1+\log_e R}{R}\times P-\frac{1+\log_e p}{r}\times p+(p-p_0),$$

assuming the final pressure of the curve of expansion if continued to be 1 lb.,

$$\frac{1+\log_e R}{R}\times P=(1+\log_e P) \text{ or } (1+\log_e R)$$

$\frac{1+\log r}{r}\times p$ will likewise be equal to $(1+\log_e p)$.

$$P_m=(1+\log_e P)-(1+\log_e p)+(p-p_0)$$

$$=\log_e P-\log_e p+p-p_0,$$

since $p_0=1$. Here again $P_m=\log_e P-\log_e p+p$.

The maximum work possible from 1 lb. of steam under these conditions,

$$=(\text{Log}_e\ \text{P} - \log_e p + p)\ 47{,}808 \text{ foot-lbs.}$$

(5) If steam expands to a pressure p, and exhausts into a receiver in which the pressure is p_1, then

$$\text{P}_m = \log_e \text{P} - \log_e p.$$

The maximum work done externally under these circumstances by 1 lb. of steam of a pressure $\text{P} = (\log_e \text{P} - \log_e p)\ 47{,}808$ foot-lbs.

B. If the steam is dry and expands adiabatically then

$$pv^{17/16} = \text{constant (69,000 foot-lbs. Rankine).}$$

Rule 33a. Then $\text{P}_m = \dfrac{17 - 16\text{R}^{-\frac{1}{16}}}{\text{R}} \times \text{P}$ . . . . . (1)

If the steam expands to pressure p and drops to the pressure p_0,

$$\text{P}_m = \left(\frac{17 - 16\text{R}^{-\frac{1}{16}}}{\text{R}}\right) \times \text{P} - \left(\frac{17 - 16r^{-\frac{1}{16}}}{r}\right) p + (p - p_0) \quad . \quad (2)$$

If $p_0 = 1$, and steam expands to p, then $\text{R} = \text{P} \div 1$ and $r = p \div 1$.

$$\text{Then } \text{P}_m = (17 - 16\text{R}^{-\frac{1}{16}}) - (17 - 16r^{-\frac{1}{16}}) + (p - 1)$$

$$= 16\,(r^{-\frac{1}{16}} - \text{R}^{-\frac{1}{16}}) + (p - 1) \quad . \quad . \quad . \quad . \quad (3)$$

When steam expands adiabatically from pressure P to 1 lb., and it condenses at pressure p, then

Maximum work done by 1 lb. of steam externally,

$$= 144v\left\{(17 - 16\text{P}^{-\frac{1}{16}}) - 1\right\}$$

$$= 2304v\,(1 - \text{P}^{-\frac{1}{16}})$$

and since $v = 333$ cubic feet.

Maximum work obtainable from 1 lb. steam,

$$= 767{,}222\,(1 - \text{P}^{-\frac{1}{16}}) \text{ foot-lbs.} \quad . \quad . \quad . \quad (4)$$

Maximum work obtainable in B thermal units,

$$= (\text{T}_1 - \text{T}_2) \times \left(1 - \frac{\text{L}_1}{\text{T}_1}\right) - \text{T}_2 \log_e \frac{\text{T}_1}{\text{T}_2} \quad . \quad . \quad (5)$$

This (Sir J. Ewing, p. 100) T_1 is the absolute temperature, and L_1 the latent heat of steam at P. T_2 the absolute temperature to which steam expands and condenses. Table VII. is calculated by (5).

Table XXI.—Steam used Expansively.

R Rate of Expansion.	1÷R Cut off.	$\frac{1+\text{hyp. log R}}{R}$ $\epsilon=1\cdot00$	$\frac{17-16R^{-\frac{1}{16}}}{R}$ $\epsilon=1\cdot0625$	R	$\frac{1}{R}$	$\frac{1+\text{hyp. log R}}{R}$ $\epsilon=1\cdot0$	$\frac{17-16R^{-\frac{1}{16}}}{R}$ $\epsilon=17/16$	R	$\frac{1}{R}$	$\frac{1+\text{hyp. log R}}{R}$ $\epsilon=1\cdot0$	$\frac{17-16R^{-\frac{1}{16}}}{R}$ $\epsilon=17/16$
50	·0200	·09825	·08942	20	·050	·1998	·186	4·00	·250	·5965	·582
48	·0209	·10150	·09248	18	·055	·2161	·200	3·63	·275	·6308	·616
46	·0218	·10498	·09580	16	·062	·2358	·220	3·33	·300	·6615	·648
44	·0227	·10875	·09920	15	·066	·2472	·230	3·00	·333	·6993	·670
42	·0238	·11282	·10317	14	·071	·2599	·242	2·86	·350	·7171	·707
40	·0250	·11724	·10735	13·33	·075	·2690	·254	2·66	·375	·7440	·733
38	·0263	·12206	·11195	13	·077	·2742	·258	2·50	·400	·7664	·756
36	·0278	·12734	·11700	12	·083	·2904	·271	2·22	·450	·8095	·800
34	·0294	·13314	·12380	11	·091	·3089	·292	2·00	·500	·8465	·840
32	·0312	·13957	·12918	10	·100	·3303	·314	1·82	·550	·8786	·874
30	·0333	·14673	·13600	9	·111	·3552	·340	1·66	·600	·9066	·900
28	·0357	·15474	·14351	8	·125	·3849	·370	1·60	625	·9187	·913
26	·0385	·16380	·15154	7	·143	·4210	·408	1·54	·650	·9292	·926
25	·0400	·16877	·15611	6·66	·150	·4347	·417	1·48	·675	·9405	·940
24	·0417	·17411	·16113	6·00	·166	·4653	·450	1·43	·700	·950	·945
23	·0435	·17982	·16661	5·71	·175	·4807	·466	1·33	·750	·965	·960
22	·0455	·18598	·17275	5·00	·200	·5218	·506	1·25	·800	·978	·976
21	·0476	·19260	·17955	4·44	·225	·5608	·540	1·18	·850	·989	·986

Table XXII.—Factors for Finding the Theoretical Mean Pressures when Clearance Volumes are taken into Account.

R Rate of Expansion.	$\frac{1}{R}$ Cut off.	Clearance Volume per cent. of Cylinder.						
		4·0	6·0	8·0	10·0	12·0	14·0	16·0
20·0	0·05	0·28	0·30	0·32	0·35	0·37	0·39	0·41
16·67	0·06	0·30	0·32	0·34	0·37	0·39	0·41	0·42
14·29	0·07	0·32	0·34	0·36	0·39	0·41	0·43	0·44
12·50	0·08	0·34	0·36	0·38	0·40	0·42	0·44	0·46
11·11	0·09	0·36	0·38	0·40	0·42	0·44	0·46	0·48
10·00	0·10	0·38	0·40	0·42	0·44	0·46	0·47	0·50
8·33	0·12	0·42	0·44	0·45	0·47	0·49	0·50	0·52
7·14	0·14	0·45	0·47	0·49	0·50	0·52	0·53	0·54
6·25	0·16	0·48	0·50	0·52	0·53	0·55	0·56	0·57
5·56	0·18	0·51	0·53	0·55	0·56	0·57	0·59	0·60
5·00	0·20	0·56	0·57	0·58	0·59	0·60	0·61	0·62
4·00	0·25	0·62	0·63	0·64	0·65	0·66	0·67	0·68
3·33	0·30	0·68	0·69	0·70	0·70	0·71	0·72	0·73
2·86	0·35	0·73	0·74	0·75	0·75	0·76	0·76	0·77
2·50	0·40	0·77	0·78	0·79	0·80	0·80	0·80	0·81
2·22	0·45	0·81	0·82	0·83	0·84	0·84	0·84	0·85
2·00	0·50	0·85	0·86	0·86	0·87	0·87	0·87	0·88
1·75	0·55	0·89	0·89	0·89	0·90	0·90	0·90	0·90
1·67	0·60	0·91	0·91	0·91	0·92	0·92	0·92	0·92

The above table is for saturated steam expanding with $pv = C$.

If it is required to ascertain the theoretical mean pressure on a cylinder having 10 per cent. clearance with steam of 120 lbs. absolute pressure cut off at 0·12 of the stroke, then

Mean pressure = 120 × 0·47 or 56·4 lbs.

If exhausting to a condenser, with a vacuum of 28 ins., then

Mean pressure = 56·4 − 1 or 55·4 lbs.

Table XXIII.—Mean Pressure of Dry Steam Expanded Adiabatically ($\epsilon = 1\frac{7}{16}$).

Rate of expansion.	Initial Pressures (absolute)									
	25 lbs.	50 lbs.	75 lbs.	100 lbs.	125 lbs.	150 lbs.	175 lbs.	200 lbs.	225 lbs.	250 lbs.
20	4·65	9·30	13·95	18·60	23·25	27·90	32·55	37·20	41·85	46·50
19	4·83	9·66	14·49	19·32	24·15	28·98	33·81	38·64	43·47	48·30
18	5·02	10·05	15·06	20·08	25·09	30·12	35·15	40·28	45·27	50·28
17	5·24	10·48	15·72	20·96	26·20	31·44	36·68	41·92	47·16	52·40
16	5·50	11·00	16·50	22·00	27·50	33·00	38·50	44·00	49·50	55·00
15	5·75	11·50	17·25	23·00	28·75	34·50	40·25	46·00	51·75	57·50
14	6·05	12·10	18·15	24·20	30·25	36·30	42·35	48·40	54·45	60·50
13	6·45	12·90	19·35	25·80	32·25	38·70	45·15	51·60	58·05	64·50
12	6·78	13·56	20·34	27·12	33·90	40·68	47·46	54·24	61·02	67·80
11	7·30	14·60	21·90	29·20	36·50	43·80	51·10	58·40	65·70	73·00
10	7·85	15·70	23·55	31·40	39·25	47·10	54·95	62·80	70·65	78·50
9	8·50	17·00	25·50	34·00	42·50	51·00	59·50	68·00	76·50	85·00
8	9·25	18·50	27·75	37·00	46·75	55·50	64·75	74·00	83·25	92·50
7	10·20	20·40	30·60	40·80	51·00	61·20	71·40	81·60	91·80	102·00
6	11·25	22·50	33·75	45·00	56·25	67·50	78·75	90·00	101·25	112·50
5	12·65	25·30	37·95	50·60	63·25	75·90	88·55	101·20	113·85	126·50
4·5	13·55	27·10	40·65	54·20	67·75	81·30	94·85	108·40	121·95	135·50
4·0	14·55	29·10	43·65	58·20	72·75	87·30	101·85	116·40	130·95	145·50
3·5	15·55	31·10	46·65	62·20	77·75	93·30	108·85	124·04	139·95	155·50
3·0	16·75	33·50	50·25	67·00	83·75	100·50	117·25	134·00	150·75	167·50
2·5	18·50	37·00	55·50	74·00	92·50	111·00	129·50	148·00	166·50	185·00
2·0	21·00	42·00	63·00	84·00	105·00	126·00	147·00	168·00	189·00	210·00
1·82	21·85	43·70	65·55	87·40	109·25	131·10	152·95	174·80	196·65	218·50
1·66	22·50	45·00	67·50	90·00	112·50	135·00	157·50	180·00	202·50	225·00
1·60	22·83	45·76	68·49	91·32	114·15	136·98	159·81	181·04	205·47	228·30
1·54	23·15	46·30	69·45	92·60	115·65	138·90	162·05	185·20	208·35	231·50
1·48	23·50	47·00	70·50	94·00	117·50	141·00	164·50	188·00	211·50	235·00

Column 4, Table XXI., gives the multipliers for *dry saturated* steam, calculated by Rankine's formula $\frac{p_m}{p_1} = \frac{17 - 16r - r^{\frac{1}{16}}}{r}$, which is based on the assumption that $pv^{1\frac{1}{16}} = \text{constant}$, and this expansion is called *adiabatic* as it takes place without any added heat as in the ordinary marine engine cylinder. At very high rates of expansion only is there very great difference between column 3 and column 4.

Absolute initial pressure should be taken at 3 to 5 lbs. below the absolute boiler pressure, and back pressure at not less than 3 lbs.; also, number of expansions is to be understood as the *nominal* number, or $\frac{\text{L.P. capacity}}{\text{H.P. capacity at cut-off}}$, without regard to clearances, or to the occurrence of release in final cylinder before end of stroke.

Then, to determine the effective p_m, when the boiler pressure is fixed, and the best number of expansions agreed upon,—multiply the p_1 by the quantity found in column 3 of Table XXI. opposite the proper number of expansions, and subtract back pressure from the product.

As values of effective mean pressures, found by aid of Table XXI., are determined without taking into consideration such disturbing elements as clearance, compression, radiation, friction of ports and passages, receiver "drop," initial condensation and subsequent re-evaporation, &c., it is necessary to allow for these in some way; but, the calculations for their values being laborious, and involving so many assumptions as to be of doubtful accuracy when finished, it is best to allow for them by multiplying by a single factor, the value of which is derived from experience.

Then $p_m \times \text{factor} = \text{R}p_m$, or referred mean pressure—(*q.v.* page 73).

Table XXIV. gives the average value of this factor for the various types of engine, and under the various conditions named:—

Table XXIV.—Ratios of Mean Pressures as in Practice to Theoretical.

Description of Engine.		Jacketed.	Unjacketed.
(1) Expansion taking place all in one cylinder; ports of average size; and ordinary slide valve driven by ordinary eccentric;—as in various auxiliary engines, . .	.	.	·76 to ·80 MEAN ·78
(2) Expansion taking place all in one cylinder; ports of average size; and ordinary slide valves driven by ordinary eccentric gear;—as in low-pressure paddle engines, .	.	.	·68 to ·75 MEAN ·71
(3) Expansion taking place in two cylinders with receiver between; ports of average size; ordinary slide valves (H.P. single-ported, and L.P. double-ported); and ordinary eccentric gear;—as in paddle engines, small screw engines, and some auxiliary engines, ...	Screw Paddle Auxly.	·67 to ·73 MEAN ·70 . .	·58 to ·68 MEAN ·63 ·55 to ·65 MEAN ·60 ·45 to ·50 MEAN ·47
(4) Expansion taking place in two cylinders, placed in same line, and exhausting direct from one to the other; ports, slide valves, and gear as in (3);—as occasionally used, ...	.	.	·71 to ·75 MEAN ·72
(5) Expansion taking place in three cylinders, placed side by side, with receivers between them; ports of average size; ordinary slide valves (say piston valve for H.P. and double-ported flat slides for M.P. and L.P.); and ordinary eccentric gear;—as used in most merchant steamers, . . .	.	·64 to ·68 MEAN ·66	·60 to ·66 MEAN ·63
(6) Engines same as (5); but ratio of H.P. to L.P. not exceeding 1:5; and running at 90 to 140 revolutions;—as in ironclads and cruisers, ...	.	·55 to ·65 MEAN ·60	.
(7) Engines same as (5); but ratio of H.P. to L.P. not exceeding 1:5; and running at 200 to 350 revolutions;—as in torpedo gunboats, &c., ...	.	.	·60 to ·67 MEAN ·63

In the case of any multiple-stage engine, where the power is divided equally amongst the cylinders, whose number is N, and where the

Table XXV.—Results of Trials of Marine Engines (Twin Screw).

Name. H.M.S.	Nature of Trials.	Cylinders. Diameters. H.P.	M.P.	L.P.	L.P.	Stroke. Inches.	Engine Room. Steam. Lbs.	Vacuum. Inches.	Revolts.	Cut-off. H.P.	Mean Pressures. H.P.	M.P.	L.P.	L.P.	Ref to L.P. Cl.	One Engine. Indicated Horse Power. H.P.	M.P.	L.P.	L.P.	Total.	Nominal Rate of Expansion.
Good Hope,	F.S.	43½	71	81½	81½	48	278	27	126	0·73	105	44·6	16·9	17·9	49 3	4,764	5,399	2,698	2,849	15,710	9·6
	R.S.						268	26·5	110·7	0·63	95·8	37·9	11·9	13·4	40·7	3,819	4,029	1,606	1,864	11,378	..
Formidable,	F.S.	31½	51½	84	..	51	245	25·5	110·2	0·70	115	44	17·9	..	50·5	2,543	2,601	2,815	..	7,959	10·1
	R.S.						240	26·5	101	..	95·2	34·4	14·3	..	40·7	1,934	1,868	2,066	..	5,868	..
Arrogant,	F.S.	26	42	68	..	39	245	26	143	0·70	112·5	47·1	20·5	..	52·8	1,685	1,841	2,100	..	5,406	9·8
	R.S.						235	27	128	0·51	86 5	35·3	15·8	..	41·2	1,159	1,233	1,449	..	3,841	13·5
Apollo,	F.D.	33½	49	74	..	39	151	26·1	137·3	0·85	51·9	32·7	13·9	..	38·8	1,236	1,669	1,621	..	4,527	5·7
	N.D.						150	28	130	..	49·4	27·3	12·1	..	36·1	1,115	1,316	1,339	..	3,770	..
Perseus,	F.S.	20½	33	54	..	27	266	24·2	214·8	0·75	..	..	..	..	52·5	992	1,263	1,268	..	3,523	9·3
	R.S.						251	25·1	197	..	..	..	..	..	42·9	707	987	942	..	2,636	..
Blanche,	F.D.	21	31	45	..	24	148	25·5	190·1	0·79	56·5	25·4	13·5	..	39·8	451	442	497	..	1,459	5·8
	N.D.						147	25·8	197·6	..	38·7	18·7	10·1	..	27·4	273	287	325	..	885	..
Dove,	F.D.	19½	30	34	34	18	206	25	385·4	0·75	77·3	45·5	17·7	17·2	47·4	808	1,127	563	547	3,045	8·7
Snapper,	F.D.	19½	28½	43	..	18	180	25	370·6	0·58	64·5	28·7	14·6	..	39·7	635	592	715	..	1,942	8·6

Table XXV.—Results of Trials of Marine Engines (Twin Screw)—*continued.*

Merchant Ships.

	Screws.																			
Victoria, T	T.S.	41	67	106	..	63	149	24	78	0·78	62·6	27·8	..	11·16	32·8	2,214	2,470	..	2,494	7,178	8·6
Tmlipa, T	S.S.	40	64	92	..	60	140	23	77	0·73	55·1	21·4	..	11·45	32·2	1,615	1,606	..	1,775	4,996	7·2
Saxonia, Q	T.S.	29	41½	59	84	54	192	25·3	78	..	73·1	41	20·9	12·08	39·0	985	1,158	1,202	1,420	4,765	..
Cazengo, T	S.S.	32	48	80	..	48	145	..	71	0·65	68·1	32	..	11·0	33·6	943	994	..	947	2,884	9·5
Othello, T	S.S.	24¾	40	70	..	48	200	28	71	0·59	88·5	36·8	..	13·0	36·7	733	794	..	903	2,430	15·2
P. Grge, T	T.S.	26	40	45	45	30	177	27	177	0·69	57·5	30·3	15·4	15·3	36·9	818	1,019	682	630	3,149	8·9
Ohio. Q .	S.S.	22½	32	45	64	42	200	27½	67·5	0·64	63·5	28·5	20·4	11	39·0	362	328	464	507	1,661	12·7
P. Edwrd, T	T.S.	19	30	48	..	24	180	25	192·3	0·68	81	33·5	..	13·7	39·6	536	554	..	577	1,667	9·4
Castro, T .	S.S.	19	33	56	..	36	200	25	91·5	0·59	87·8	33·5	..	10·9	32·5	417	470	..	445	1,333	14·7
Colchstr, T	T.S.	14½	23	27	27	33	180	26·5	145	0·75	88	38·0	18·4	18·2	44·6	331	382	255	252	1,220	9·3
Molopo, TQ	S.S.	13	22	36	..	24	200	23·6	112	0·60	96·7	38·7	..	13·1	40·0	174	199	..	181	554	12·8
Polo, Q .	S.S.	12¾	13½	27	39	24	200	25	98	0·62	75	34·3	15·3	11·85	34·8	113	110	104	168	495	15·1
Ptn, Q . .	S.S.	24	34½	49	71	48	220	28·0	65	..	80·3	32·1	16·0	9·12	33·5	572	476	472	568	2,088	..
Vsr, Q . .	S.S.	26½	37½	59	77	57	230	27·5	75	..	80·6	44·7	18·4	10·6	41·3	964	1,068	1,059	1,068	4,159	..
Tk, Q . .	S.S.	20	28	39½	57	33	212	25·0	140	0·63	73·5	37·0	19·8	10·0	40·0	540	532	565	596	2,233	12·9
Hl, Q . .	S.S.	22½	31½	45	66	51	215	25·5	76	0·57	74·0	35·5	18·3	8·3	33·3	570	542	566	550	2,228	15·1
Lru, T . .	S.S.	22	35	..	53	33	205	26·0	114	..	81·2	33·2	..	9·5	31·6	350	400	..	330	1,080	..
Rw, T . .	S.S.	25	40	..	68	45	180	27·2	61	..	68·7	27·1	..	10·15	30·6	500	502	..	545	1,547	..

referred mean pressure is Rp_m, the mean pressures in the various cylinders are as follows:—

$$\text{L.P. mean pressure} = \frac{Rp_m}{N}.$$

$$\text{M.P. ,, } = \frac{Rp_m}{N} \times \frac{\text{L.P. area}}{\text{M.P. area}}.$$

$$\text{H.P. ,, } = \frac{Rp_m}{N} \times \frac{\text{L.P. area}}{\text{H.P. area}}.$$

PISTON SPEED.

Usually reckoned in feet per minute.

Mean value = 2 × stroke in feet × revolutions per minute.

Experience with Naval machinery has shown that the largest pistons (*i.e.* up to 100 inches diameter) may be safely run at speeds up to 850 and 1000 feet per minute.

With the lighter machinery of torpedo boats and destroyers speeds of 1000 to 1200 feet per minute are quite satisfactory.

The standard practice in cargo steamers is very fairly represented by the expression $144\sqrt[3]{S}$, where S is the stroke in inches. (North-East Coast rule—*see* page 5).

In large passenger and mail steamers the speed is now commonly 700 to 950 feet per minute on service, or, $220\sqrt[3]{S}$.

In the case of paddle engines, and especially of oscillating paddle engines, the available range of piston speeds is very closely limited by the other conditions of the case, and has not often in practice been exceeded,—

For vertical oscillating engines, . .	450 feet per min.	or $110\sqrt[3]{S}$.
,, inclined ,, . .	520 ,,	or $118\sqrt[3]{S}$.
,, diagonal engines with connecting rods,	550 to 700 ,,	or $130\sqrt[3]{S}$.

As the powers obtained from a given weight and bulk of engine vary almost directly as the revolutions, the efforts of engineers are constantly directed to obtain higher rates, without undue increase of wear and tear, and there is therefore every probability of advance in this direction.

By increasing piston speed a perceptible reduction can also be made in the percentage of heat lost by radiation.

There are two ways of getting increased piston speed, viz.:—

(1) Increase length of stroke and let revolutions remain same;

(2) Increase revolutions and let length of stroke remain same; but there is this important difference in the result obtained,—

In the first case, the cost, bulk, and weight of engines all increase with the piston speed; whilst in the second case there is practically no increase under any of these heads; and it is therefore in this second direction that the designer must go when limited as regards space and weight.

It must not be forgotten, however, that, for a given cylinder capacity, the long stroke and small diameter cylinder has a distinct advantage over its rival, in its smaller piston area, and consequently reduced loads and stresses, besides smaller *volume* clearances.

The chief limits to increase of piston speed (and of revolutions) are, —the size of ports and consequent size and weight of valves, and the inertia of reciprocating parts.

REVOLUTIONS.

In the case of paddle engines, the revolutions are, of course, strictly governed by the diameter of wheel and speed of vessel; increased power must therefore be obtained either by increased stroke, pressure, or diameter of cylinders.

Rate of Revolution at full speed of reciprocating engines varies roughly as the $\sqrt{\text{N.H.P.}}$ *v.* Rule 7.

Rule 34. Revs. per min. $= Q \div \sqrt{\text{N.H.P.}}$
For the ordinary merchant steamers, $Q = 1200$.
For express steamships, $Q = 1800$.

Rule 34a. For paddle wheelers. Revs. per min. $= 280 \div \sqrt[3]{\text{N.H.P.}}$

Rule 34b. For turbine-driven screws their rate of revolution = speed in knots $\times$ F.
For single-geared, Naval, and cross-channel ships, $F = 11$.
For double-geared ships, $F = 6{\cdot}5$.
The following Table gives some examples of various ships' rates of revolutions when at full speed :—

Table XXVI.—Revolutions, Rates of, in Practice.

Class of Ship.	Propelling Machinery.	Horse-Power per Screw.	Revolutions P.M.	
			Engines.	Screws.
Battleships and Cruisers	Turbines, Direct Drive	19,000	305	305
Do. do.	Do. Single Reduction	37,500	1470	206
Cruisers, First Class	Do. Direct Drive	12,000	260	260
Do. do.	Do. Single Reduction	26,000	2580	330
Do. Light	Do. Direct Drive	6,250	500	500
Do. do.	Do. Single Reduction	18,000	3200	420
T.B.D. Squadron Leaders	Do. Direct Drive	16,000	690	690
Do. do. do.	Do. Single Reduction	22,000	3000	350
Ocean Express	Steam Reciprocators	22,500	82	82
Do. do.	Turbines, Direct Drive	28,000	188	188
Do. do.	Do. Single Reduction	5,500	1700	137
Do. do.	Do. Double Reduction	6,250	3200	80
Passenger Cargo	Steam Reciprocators	4,600	95	95
Do. do.	Oil do.	2,250	125	125
Do. do.	Turbines, Single Reduction	4,150	1000	96
Do. do.	Do. Double Reduction	3,750	3500	95
Cross-Channel Express	Do. Direct Drive	4,600	670	670
Do. do. do.	Do. Single Reduction	8,000	2800	400
Do. do. Large	Do. do. do.	7,700	1984	287
Cargo Single Screw	Steam Reciprocators	1,250	78	78
Do. do. do.	Oil do.	1,600	115	115
Do. do. do.	Turbines Geared	1,250	3500 H.P.	80

Table XXVII.—Relation between Stroke and Revolutions, Cargo Steamers.

Stroke in inches.	Revolutions per minute.	Stroke in inches.	Revolutions per minute.	Stroke in inches.	Revolutions per minute.
18	125	33	84	48	65
21	113	36	79	51	63
24	103	39	75	54	61
27	96	42	71	57	58
30	89	45	68	60	56

STROKE OF PISTON

The following are the lengths of stroke usual in the various classes of vessels named :—

Table XXVIII.—Stroke of Piston.

Battle Ships and First Class Cruisers, . .	45 to 51 inches.
Second Class Cruisers, ...	36 to 42 ,,
Scouts and Third Class Cruisers, . . .	24 to 30 ,,
Torpedo Gunboats and Destroyers, . . .	18 to 21 ,,
Torpedo Boats, ...	9 to 18 ,,
Large Mail Steamers, ...	60 to 78 ,,
Fast Passenger Steamers, ...	30 to 48 ,,
Ordinary Merchant Steamers, . . .	18 to 60 ,,
Paddle Steamers, ...	30 to 102 ,,

Engines of the overhead beam type, used in paddle vessels, have sometimes had strokes of 10 feet and even 12 feet.

In ordinary cargo steamers the length of stroke is usually about ·65 × diameter of L.P. cylinder. (*See* Table I., page 5).

TO CALCULATE DIAMETER OF CYLINDER FOR A GIVEN POWER.

The following rules apply to L.P. cylinders only,—the sizes of M.P. and H.P. cylinders being supposed to be fixed from those of the L.P. by the rules given in section on "Ratios of Cylinders," pages 57–65.

For rapid calculations, use the formula for Estimated Horse Power, Rule 1 ; the result will usually be within 5 per cent.

Rule 1. $$\text{E.H.P.} = \frac{D^2 \times \sqrt{p} \times R \times S}{Z}$$

where D = diameter of L.P. cylinder in inches.
p = absolute boiler pressure.
R = revolutions per minute.
S = stroke in feet.
Z a factor (*v.* page 8).

If the size of cylinder for a given **Indicated Horse-power** is required, let S stand for piston speed in feet per minute, and Rp_m for the referred mean pressure in pounds per square inch (calculated by rules given on pages 67, 68); then,—

Rule 35. **Area of L.P. piston** $= \frac{\text{I.H.P.} \times 33{,}000}{Rp_m \times S}$;

or, if piston speed is required,—

Rule 36. **Piston speed** $= \frac{\text{I.H.P.} \times 33{,}000}{\text{L.P. area} \times Rp_m}$;

or, again, if referred mean pressure is required,—

Rule 37. Referred mean pressure $Rp_m = \frac{\text{I.H.P.} \times 33{,}000}{\text{L.P. area} \times S}$;

and lastly, if I.H.P. is required,—

Rule 38. $$\text{I.H.P.} = \frac{\text{L.P. area} \times S \times Rp_m}{33{,}000}.$$

CYLINDER PORTS, PIPES, AND PASSAGES.

In fixing the sizes of cylinder ports, &c., it is necessary—as in so many other cases—to discover the best compromise. By increasing the port areas, for instance, "wire-drawing" is diminished, a freer exhaust is obtained, and the resulting indicator diagram is fuller; but, at the same time, the size and weight of cylinder, of slide-valve, and of valve-gear, as also the clearance volume, are increased, the loss from which may possibly balance the gain.

The following figures give speeds of steam usual in good triple and quadruple engines; but it must be understood that in dealing with very high piston speeds (say over 900 feet per minute), it is not always either possible or advisable to give such large areas:—

Table XXIX.—Speeds of Steam-flow through Ports, &c.

Main steam pipe, 8100 ft. per minute; then—

$$\text{Diameter} = \frac{\text{Dia. of H.P. cyl.}}{90} \times \sqrt{\text{Mean piston speed.}}$$

Mean of maximum valve openings,	H.P. — 7,500 ft. per min. M.P. — 9,000 ,, L.P. — 12,000 ,,		
Ports (during exhaust),	H.P. — 5,800 ,, M.P. — 7,200 ,, L.P. — 8,600 ,,	Nearly equivalent to 40, 50, and 60 c. ft. of cylinder per sq. inch of port per minute.	
Exhaust pipe or passage from one cylinder to next or to condenser,	H.P. — 4,500 ,, M.P. — 5,500 ,, L.P. — 7,000 ,,		
Ports (during exhaust) in light, high-speed engines,	H.P. — 5,800 ,, M.P. — 8,600 ,, L.P. — 11,500 ,,	Nearly equivalent to 40, 60, and 80 c. ft. of cylinder per sq. inch of port per minute.	

For **Two-stage Compound** engines use the above figures, only omitting those referring to M.P. cylinders, and for quadruples the mean speeds between H.P. and M.P. and M.P. and L.P.

The following Table is based on the above figures, and gives the proportions at a glance,—where A is the area of the cylinder :—

Table XXX.—Ratios for Ports, Pipes, &c., of Cylinders.

Piston speed, feet per minute.	H.P.A. ÷ Area of main steam.	H.P.A. ÷ Mean area of H.P. max. openings.	M.P.A. ÷ Mean area of M.P. max. openings.	L.P.A. ÷ Mean area of L.P. max. openings.	H.P.A. ÷ Area of H.P. ports.	M.P.A. ÷ Area of M.P. ports.	L.P.A. ÷ Area of L.P. ports.	H.P.A. ÷ Area of pipe from H.P. to M.P.	M.P.A. ÷ Area of pipe from M.P. to L.P.	L.P.A. ÷ Area of main eduction.
200	40·0	37·5	45·0	60·0	29·0	36·0	43·0	22·5	27·5	35·0
250	32·0	30·0	36·0	48·0	23·2	28·8	34·4	18·0	22·0	28·0
300	26·6	25·0	30·0	40·0	19·3	24·0	28·6	15·0	18·3	23·3
350	22·8	21·4	25·7	34·3	16·6	20·6	24·6	12·8	15·7	20·0
400	20·0	18·7	22·5	30·0	14·5	18·0	21·5	11·2	13·7	17·5
450	17·7	16·6	20·0	26·6	12·9	16·0	19·1	10·0	12·2	15·5
500	16·0	15·0	18·0	24·0	11·6	14·4	17·2	9·0	11·0	14·0
550	14·5	13·6	16·3	21·8	10·5	13·1	15·6	8·2	10·0	12·7
600	13·3	12·5	15·0	20·0	9·6	12·0	14·3	7·5	9·1	11·6
650	12·3	11·5	13·9	18·4	8·9	11·1	13·2	6·9	8·4	10·7
700	11·4	10·7	12·8	17·1	8·3	10·3	12·3	6·4	7·8	10·0
750	10·6	10·0	12·0	16·0	7·7	9·6	11·5	6·0	7·3	9·3
800	10·0	9·4	11·2	15·0	7·2	9·0	10·7	5·6	6·9	8·7
850	9·4	8·8	10·6	14·1	6·8	8·5	10·1	5·3	6·5	8·2
900	8·9	8·3	10·0	13·3	6·4	8·0	9·5	5·0	6·1	7·8
950	8·4	7·9	9·5	12·6	6·1	7·6	9·0	4·7	5·8	7·4
1000	8·0	7·5	9·0	12·0	5·8	7·2	8·6	4·5	5·5	7·0

To determine the area of any port, pipe, passage, &c., for any given speed of piston,—Divide the area of the cylinder by the number found in the proper column opposite the given piston speed, and quotient will be area required.

GENERAL REMARKS ON STRENGTHS, &c.

The dimensions and proportions given for the various details of engines, in this and succeeding sections, are in all cases such as experience has shown to give satisfactory results, both as regards stiffness, strength, and durability. Where weight is not an important consideration the various parts may be made heavier, but with good material this is unnecessary; on the other hand, they may be, and are sometimes, made lighter,—as in very high-speed Naval vessels,—but in such cases the increased risk of breakdown, and wear and tear when working at full power, are understood, and accepted as the price paid for reduction in weight. Such ships, however, do not run often or for long periods at full power.

CYLINDER BARRELS.

Cylinder barrels should be made of good sound cast-iron, at least twice melted, and of the thicknesses given by the following rules:—

Rule 39. $T = \frac{D}{6500}(p+50)+\cdot2$; for cylinders fitted with liners.

Rule 39a. $T = \frac{D}{6500}(p+50)+\cdot4$; for cylinders without liners.

Where T is thickness of barrel, and D diameter of cylinder, both in inches; and p is maximum pressure in cylinder,—the values of which are assumed to be as follows:—

For H.P. cylinders of Triple and Quadruple engines, p = boiler pressure.

H.P.	,,	Compound	,,	p = boiler pressure + 20.
M.P.	,,	Triple	,,	$p = \cdot6$ boiler pressure.
M.P.$_1$	,,	Quadruple	,,	$p = \cdot7 \times$,,
M.P.$_2$	,,	,,	,,	$p = \cdot45 \times$,,
L.P.	,,	Compound	,,	$p = \cdot5 \times$ (boiler pressure + 20).
L.P.	,,	Triple	,,	$p = \cdot37 \times$ boiler pressure.
L.P.	,,	Quadruple	,,	$p = \cdot3 \times$,,

It is also assumed, as previously stated, that compound engines work at pressure between 70 lbs. and 120 lbs.; Triple engines between 120 lbs. and 180 lbs.; and Quadruple engines between 180 lbs. and 240 lbs.

For working pressure of 160 lbs. the formula may be reduced to the forms given at the heads of the columns in the following Table, which is calculated in accordance with them, for 180 lbs.:—

Table XXXI.—Cylinder Barrels of Triple Engines (180 lbs.).

Dia. of Cylinder in ins.	Thickness in inches, WITH LINERS.						Thick. in inches, WITHOUT LINERS.					
	H.P. $\frac{D}{28\cdot5}+\cdot2$		M.P. $\frac{D}{41}+\cdot2$		L.P. $\frac{D}{55}+\cdot2$		H.P $\frac{D}{28\cdot5}+\cdot4$		M.P. $\frac{D}{41}+\cdot4$		L.P. $\frac{D}{55}+\cdot4$	
10							·81	13/16	·75	3/4	·58	5/8
15	·90	15/16					·95	15/16	·81	13/16	·67	11/16
20	1·07	1 1/16					1·10	1 1/8	·89	15/16	·76	15/16
25	1·25	1 1/4	·81	13/16			1·27	1 1/4	1·01	1	·85	7/8
30	1·42	1 7/16	·93	15/16			1·45	1 7/16	1·13	1 1/8	·94	1
35	1·60	1 5/8	1·05	1 1/16	·83	13/16	1·62	1 5/8	1·25	1 1/4	1·03	1 1/16
40	1·77	1 3/4	1·17	1 3/16	·93	15/16	1·80	1 13/16	1·37	1 3/8	1·13	1 1/8
45	1·95	1 15/16	1·29	1 5/16	1·02	1 1/16	1·97	2	1·49	1 1/2	1·22	1 1/4
50	2·12	2 1/8	1·42	1 7/16	1·11	1 1/8	2·15	2 1/8	1·62	1 5/8	1·31	1 5/16
55	2·30	2 5/16	1·54	1 9/16	1·20	1 1/4	2·32	2 5/16	1·74	1 3/4	1·40	1 7/16
60			1·66	1 11/16	1·29	1 5/16	2·50	2 1/2	1·86	1 7/8	1·49	1 1/2
65			1·78	1 13/16	1·38	1 3/8			1·98	2	1·58	1 9/16
70			1·90	1 15/16	1·47	1 1/2			2·10	2 1/8	1·67	1 11/16
75			2·03	2 1/16	1·56	1 9/16			2·23	2 1/4	1·76	1 3/4
80			2·15	2 3/16	1·65	1 11/16			2·35	2 3/8	1·85	1 7/8
85					1·74	1 3/4					1·94	1 15/16
90					1·83	1 7/8					2·03	2 1/16
95					1·93	1 15/16					2·13	2 1/8
100					2·02	2 1/16					2·22	2 1/4
105					2·11	2 1/8					2·31	2 5/16
110					2·20	2 1/4					2·40	2 7/16

The following general rule will be useful where pressures of over 200 lbs. per square inch are in question, as it has a wider range than those given above :—

Rule 40. —For the thickness of any cylinder of a compound, triple, or quadruple engine.

$$\text{Thickness} = \frac{D}{6400}(p+80) \times \frac{1}{\sqrt{r}} + \begin{cases} \cdot2 \text{ with liner.} \\ \cdot4 \text{ without liner.} \end{cases}$$

where D is diameter of cylinder in inches, p working pressure at engines, and r has the values given below.

Working Pressure.	Values of r.				
	H.P.	I.P. Triple.	1st I.P. Quadruple.	2nd I.P. Quadruple.	L.P.
100 lbs.	1	...	...	...	2·5
160 ,,	1	2·3	...	...	4
200 ,,	1	2·5	1·7	3	5·55
250 ,,	1	2·75	1·75	3·45	7·5

Table XXXII.—Cylinder Barrels of 2-Stage Compound Engines (100 lbs.).

Dia. of Cylinder in ins.	Thickness in inches, WITH LINERS. H.P. $\frac{D}{35\cdot3}+\cdot2$		Thickness in inches, WITH LINERS. L.P. $\frac{D}{55}+\cdot2$		Thick. in inches, WITHOUT LINERS. H.P. $\frac{D}{35\cdot3}+\cdot4$		Thick. in inches, WITHOUT LINERS. L.P. $\frac{D}{55}+\cdot4$	
10					·68	11/16	·58	5/8
15					·82	15/16	·67	11/16
20	·76	3/4			·96	1	·76	13/16
25	·91	15/16			1·11	1 1/8	·85	7/8
30	1·05	1 1/16			1·25	1 1/4	·94	1
35	1·19	1 3/16	·83	13/16	1·39	1 3/8	1·03	1 1/16
40	1·33	1 5/16	·93	15/16	1·53	1 1/2	1·13	1 1/8
45	1·47	1 1/2	1·02	1 1/16	1·67	1 11/16	1·22	1 1/4
50	1·61	1 5/8	1·11	1 1/8	1·81	1 13/16	1·31	1 5/16
55	1·75	1 3/4	1·20	1 1/4	1·95	1 15/16	1·40	1 7/16
60	1·90	1 7/8	1·29	1 5/16	2·10	2 1/8	1·49	1 1/2
65	2·04	2 1/16	1·38	1 3/8	2·24	2 1/4	1·58	1 9/16
70			1·47	1 1/2			1·67	1 11/16
75			1·56	1 9/16			1·76	1 3/4
80			1·65	1 11/16			1·85	1 7/8
85			1·74	1 3/4			1·94	1 15/16
90			1·83	1 7/8			2·03	2 1/16
95			1·93	1 15/16			2·13	2 1/8
100			2·02	2 1/16			2·22	2 1/4
105			2·11	2 1/8			2·31	2 5/16
110			2·20	2 1/4			2·40	2 7/16

For the very light machinery of Torpedo boats and Destroyers, the thicknesses of cylinders for triple engines working at pressures of 220 to 250 lbs. may be as shown in the following Table :—

Table XXXIII.—Cylinder Barrels of Torpedo Boats and Destroyers.

Diameter of cylinder in inches.	H.P. Thickness in inches. $\frac{D}{40}+\cdot34$.		M.P. Thickness in inches. $\frac{D}{80}+\cdot4$.		L.P. Thickness in inches. $\frac{D}{120}+\cdot42$.	
10	·59	9/16	...	...	...	...
15	·71	11/16	·58	9/16	...	...
20	·84	13/16	·64	5/8	·58	9/16
25	·96	15/16	·71	11/16	·62	5/8
30	1·09	1 1/16	·77	3/4	·66	11/16
35	1·21	1 5/16	·83	13/16	·71	11/16
40			·90	7/8	·75	3/4
45			·96	15/16	·79	13/16
50					·84	13/16
55					·88	7/8

The mixture of iron for cylinders of this type must, of course, be of a very special character, and should contain a good proportion of best cold-blast iron.

For thicknesses of cylinder barrels for oscillating engines see page 99.

The barrels of all cylinders, but especially the L.P., are improved by the addition of external stiffening ribs or rings; these may have a thickness of 1·5 × thickness of barrel, and may stand ·75 × thickness of barrel above the surface, whilst they may be pitched about 12 × thickness of barrel apart.

CYLINDER LINERS.

Cast-iron cylinder liners should be of thickness given by the rule,—

Rule 41. $$T = \cdot 8 \left\{ \frac{D}{6500}(p+50) \right\} + \cdot 35$$

where T is thickness, and D diameter, both in inches, and p has the same values as given in section on cylinder barrels, above.

The following Table is calculated by means of the above formula, for liners of triple engines working at 160 lbs. pressure :—

Table XXXIV.—Cast-iron Cylinder Liners (180 lbs.).

Diameter in inches.	Thickness in inches.						Diameter in inches.	Thickness in inches.					
	H.P.		M.P.		L.P.			H.P.		M.P.		L.P.	
10	...	...	...	...	...	...	65	...	...	1·62	1 5/8	1·29	1 5/16
15	...	...	...	...	...	...	70	...	...	1·71	1 11/16	1·36	1 3/8
20	0·91	1 5/16	...	...	...	...	75	...	...	1·81	1 13/16	1·44	1 7/16
25	1·05	1 1/16	0·83	1 3/16	...	...	80	...	...	1·91	1 15/16	1·51	1 1/2
30	1·19	1 3/16	0·93	1 5/16	...	...	85	...	...	...	...	1·59	1 9/16
35	1·33	1 7/16	1·03	1	0·85	7/8	90	...	...	...	...	1·66	1 5/8
40	1·47	1 1/2	1·13	1 1/8	0·92	1 5/16	95	...	...	...	...	1·73	1 3/4
45	1·61	1 5/8	1·22	1 1/4	1·00	1	100	...	...	...	...	1·80	1 15/16
50	1·75	1 3/4	1·32	1 5/16	1·08	1 1/16	105	...	...	...	...	1·88	1 7/8
55	1·89	1 7/8	1·42	1 7/16	1·15	1 1/8	110	...	...	...	...	1·95	1 15/16
60	2·03	2	1·52	1 1/2	1·22	1 1/4	...	...	...	...	...	...	...

When of forged steel, cylinder liners should be of thickness given by

Rule 42. $$T = \cdot 65 \left\{ \frac{D}{6500}(p+50) \right\} + \cdot 3$$

where all symbols have same meanings as above.

Table XXXV. is calculated by means of this formula, for liners of triple engines working at 180 lbs. pressure.

Table XXXV.—Forged steel Cylinder Liners (180 lbs.).

Diam. of cylinder in inches.	Thickness in inches.					
	H.P.		M.P.		L.P.	
20	·75	¾				
25	·87	⅞				
30	·98	1	·77	13/16		
35	1·10	1⅛	·85	⅞		
40	1·21	1¼	·93	15/16		
45	1·33	15/16	1·01	1	·83	⅞
50	1·44	17/16	1·09	1⅛	·89	15/16
55	1·55	19/16	1·17	15/16	·95	1
60	1·66	1 11/16	1·25	1¼	1·01	1 1/16
65			1·33	15/16	1·07	1 1/16
70			1·41	17/16	1·12	1⅛
75			1·49	1½	1·18	1 3/16
80			1·57	19/16	1·24	1¼
85					1·30	15/16
90					1·36	1⅜
95					1·42	17/16
100					1·48	1½
105					1·54	19/16
110					1·60	1⅝

For liners of horizontal cylinders the above thicknesses may be increased by 1/16-inch.

For steel liners the piston packing rings should be of hard bronze, as it has been found to work more satisfactorily than cast iron with the forged steel.

For cylinder liners of engines working at pressures over 200 lbs. per square inch, substitute $\frac{D}{7000}(p+80)\times\frac{1}{\sqrt{r}}$ for $\frac{D}{6500}(p+50)$ in Rules 41 and 42.

CYLINDER ENDS AND COVERS.

In most engines the cylinder end has, in addition to supporting the uniformly distributed load due to the steam pressure, to take the more or less locally applied pull of the frames or columns, and distribute it to the barrel; and the various ribs or webs must therefore be carefully arranged to effect this.

When the L.P. diameter is over 70 inches, it is desirable to make the end double, and this must of course be done for any size when steam jackets are required.

The double bottom adds considerably to the weight of a cylinder, as the inner metal must still be made strong enough to stand all local shock and strain; and it increases the risk of unsoundness in the casting.

In Naval work the double bottom is frequently dispensed with, on account of its weight; but owing to the use of the conical or "dished"

steel piston, the cylinder end can also be dished, and is thus given such additional structural strength that L.P. cylinders are commonly made with single ends up to 90 inches diameter.

Table XXXVI. gives the thicknesses of single and double cylinder ends for triple engines working at 180 lbs. pressure, the following conditions being assumed:—

(*a*) The over-all depth or thickness of a double end is not less than five times that of the metal given in the table.

(*b*) In a single end, the total depth of the central ring (forming the hole for boring-bar) is not less than $5\frac{1}{2}$ times the thickness of end given in the table.

(*c*) The number, strengths, and positions of webs, &c., are as customary in "average" practice for two, three, or four columns or frame attachments.

As much depends on (*c*) the Table and rules from which it is calculated must be used with judgment, and checked by the rule for flat surfaces given on page 90.

The rules from which Table XXXVI. is calculated are,—

Rule 43. **Thickness of single cylinder end** $= \cdot 85 \times f$.

Rule 43a. **Thickness of double cylinder end** $= \cdot 7 = f$,

where f is thickness of barrel (columns 8, 10, and 12, Table XXXI.) + ·25 in.

Table XXXVI.—Cylinder Ends of Triple Compound Engines.

Diam. of cylin. in inches	H.P.—Thick. in inches.				M.P.—Thick. in inches.				L.P.—Thick. in inches.			
	Single end.		Double end.		Single end.		Double end.		Single end.		Double end.	
10	·85	⅞	·70		·76	¾	·63		·70	¾	·58	
15	1·00	1	·83		·87	⅞	·71		·78	13/16	·64	
20	1·15	1 1/16	·94		·97	1	·80		·86	⅞	·71	
25	1·29	1 5/16	1·06		1·07	1 1/16	·88		·94	15/16	·77	
30	1·44	1 7/16	1·19	1 3/16	1·17	1 3/16	·96		1·01	1	·83	
35	1·59	1 9/16	1·31	1 5/16	1·27	1¼	1·05		1·09	1⅛	·89	
40	1·74	1¾	1·43	1 7/16	1·38	1⅜	1·13	1⅛	1·17	1 3/16	·96	
45	1·89	1⅞	1·55	1 9/16	1·48	1½	1·22	1¼	1·25	1¼	1·03	
50	2·04	2 1/16	1·67	1 11/16	1·59	1⅝	1·31	1 5/16	1·32	1 5/16	1·09	
55	2·19	2 5/16	1·80	1 13/16	1·69	1 11/16	1·40	1 7/16	1·40	1⅜	1·15	
60	2·34	2 5/16	1·92	1 15/16	1·79	1 13/16	1·48	1½	1·48	1½	1·22	
65					1·89	1⅞	1·54	1 9/16	1·56	1 9/16	1·28	1 5/16
70					2·00	2	1·60	1⅝	1·63	1⅝	1·35	1⅜
75					2·11	2 1/16	1·66	1 11/16	1·71	1¾	1·41	1 7/16
80					2·21	2 3/16	1·82	1 15/16	1·78	1 13/16	1·47	1½
85									1·86	1⅞	1·53	1 9/16
90									1·94	1 15/16	1·60	1⅝
95									2·02	2	1·66	1 11/16
100									2·10	2⅛	1·73	1¾
105									2·18	2 3/16	1·79	1 13/16
110									2·25	2¼	1·85	1⅞

N.B.—The stiffening by webs and gussets must be made to suit the various initial pressures.

The practice of making the inner metal of double bottoms and covers a little heavier than the outer in large cylinders is reasonable, the inner metal may be made $\frac{1}{16}$-inch thicker than shown in Table, and the outer $\frac{1}{16}$-inch lighter than table.

Covers are now often made single and well ribbed both in cast iron and steel; the ribs of the larger covers have bulb or T edges. Single covers should be of thickness given by,—

Rule 44. **Thickness of single cylinder cover** $= \cdot 77 \times f$;

and for double covers,—

Rule 44a. **Thickness of double cylinder cover** $= \cdot 65 \times f$;

similar general conditions being assumed, as in the case of cylinder ends.

A sufficiently close approximation to these thicknesses, for triple engines working at 180 lbs. pressure, may be obtained by multiplying the thicknesses given in Table XXXVI. for cylinder ends by ·95.

In large covers and bottoms there should be no lack of strength in the central ring which surrounds the stuffing-box or manhole in the one case, and the boring hole in the other; it should have the same thickness as the metal of the cover or end, and good round corners formed by its junction with the metals of the bottom or cover, and of the various webs.

Very light cylinders, of the type referred to in Table XXXIII., are better made with loose covers for both ends, the thicknesses of which should be as follows :—

Rule 45. **Thickness of end covers** $= \cdot 5 \times f_1$;

where f_1 is the thickness of cylinder barrel (from Table XXXIII.) + ·25 inch.

They should be made of the same special iron as the barrels, and should each have from 6 to 12 deep stiffening ribs, the mean thickness of which (they should have $\frac{1}{16}$-inch taper for moulding) may be $\frac{1}{16}$-inch less than that of the cover.

CYLINDER VALVE BOXES, PORTS, &c.

The casings, ports, passages, &c., for piston slide-valves should be made of thicknesses given by,—

Rule 46. **Thickness of piston valve casing** $= \cdot 8 \times f$;

and when flat slide-valves are used the thicknesses of casings may be,—

Rule 47. **Thickness of valve casing** $= \cdot 7 \times f$;

f having, in both cases, the same value as above.

It will be noted that the latter value ($\cdot 7 \times f$) is the same as for metal of double ends, the thicknesses of which (for triple engines working at 180 lbs. pressure) are given in Table XXXVI.

It must be remembered that in the case of valve casings, as in that of cylinder ends, a very great deal depends on the number, size, and arrangement of stiffening webs, and the following rule should therefore be used in conjunction with the Table :—

Rule for flat surfaces.—All flat surfaces should be stiffened by webs or stays of some form, whose distance apart should not exceed,—

Rule 48. **Pitch in inches** $= \sqrt{\frac{t^2 \times C}{p}}$;

where t is the thickness in 16ths of an inch, p the pressure in pounds per square inch on the surface, and C equals 54 for cast iron, and 120 for cast steel. *See* also Table CXXXIX.

These webs should be of the same thickness as the flat surface, and their depth at least 2·5 times the thickness.

When flat slide-valves are used the thickness of the metal which separates the steam ports from the steam chest or receiver, and from the exhaust port, should be in accordance with the following rule,—it being understood that the ports of large cylinders are so divided by webs that no section or compartment exceeds 20 inches in width :—

Rule 49. **Thickness of steam passage metal** $= \cdot 65 \times f$,

where f has the same value as before.

These thicknesses (for triple engines working at 180 lbs. pressure) may be obtained with sufficient accuracy by multiplying the figures found in columns 8 and 12 of Table XXXVI. by ·95.

The slide-valve faces of cylinders, when a false face is fitted, should be made of thickness given by the rule,—

Rule 50. **Thickness of cylinder valve-face (when a false face is fitted)** $= \cdot 8 \times f$,

and when no false face is used the rule becomes,—

Rule 50a. **Thickness of cylinder valve-face (when no false face is used)** $= \cdot 9 \times f$.

False faces of hard cast iron may have a thickness,—

Rule 51. **Thickness of cast-iron false face** $= \cdot 75 \times f$.

CAST-STEEL CYLINDER COVERS.

For a flat plate secured round the edge after the manner of a cylinder cover, but yet not quite encastré, the formula may be written,—

$$t = \sqrt{\frac{3r^2p}{4f}} \quad . \quad . \quad . \quad (a)$$

where t is the thickness in inches; r the radius of the cylinder,—also in inches; p the maximum pressure on cover in lbs. per square inch above atmosphere; and f the greatest permissible stress per square inch on the material. When a suitable value of f has been inserted, and a small constant quantity (necessary for practical reasons) added, this becomes,—

Rule 52. $$t = \sqrt{\frac{3r^2p}{160{,}000}} + \cdot 25,$$

which gives the thickness (very nearly) of steel cylinder covers of the single type, ribbed as directed below, and coned to suit steel pistons, made in accordance with formulæ and Table given on pages 106, 108.

For steel covers of triple engines, working at 180 lbs. pressure, this formula (after reduction and further slight adjustment of constant) takes the forms given at heads of columns in following Table:—

Table XXXVII.—Cast-Steel Cylinder Covers.

Dia. of cylinder in inches.	H.P.			M.P.			L.P.		
	Thick. in inches $\frac{D}{36}+\cdot 24$.		No. of ribs $\frac{D+20}{6}$	Thick. in inches $\frac{D}{56}+\cdot 34$.		No. of ribs $\frac{D+20}{7\cdot 5}$	Thick. in inches $\frac{D}{100}+\cdot 42$.		No. of ribs $\frac{D+20}{9}$
10	·52	9/16	5	·52	9/16	4	·52	9/16	4
20	·79	13/16	7	·70	3/4	6	·62	5/8	5
30	1·07	1 1/16	9	·87	7/8	7	·72	3/4	6
40	1·35	1 3/8	10	1·05	1 1/16	8	·82	7/8	7
50	1·63	1 5/8	12	1·23	1 1/4	10	·92	15/16	8
60	1·91	1 15/16	14	1·41	1 7/16	11	1·02	1 1/16	9
70	...	...	...	1·59	1 5/8	12	1·12	1 1/8	10
80	...	...	...	1·77	1 13/16	14	1·22	1 1/4	12
90	...	...	...	...	...	...	1·32	1 3/8	13
100	...	...	...	...	...	...	1·42	1 7/16	14
110	...	...	...	...	...	...	1·52	1 9/16	15

For higher pressures multiply the thickness given by $\sqrt{\frac{p}{180}}$, p being the pressure.

The thickness of the ribs may be ·9 × thickness of cover.

The steeper cone of the H.P. cover undoubtedly gives it considerable additional strength, but its greater liability to shock and strain from the presence of water may be considered as almost balancing the account.

CAST-IRON VALVE BOX COVERS.

Where these are circular, as in the case of covers for piston-valve boxes, a modification of the formula (a) given above may be again employed; the nature of the material renders the addition of a larger constant quantity necessary, and provision must be made for stiffness in L.P. covers by adding a constant to the pressure also. The formula may then be written,—

Rule 53. $$t = \sqrt{\frac{3 \times r^2 \times (p+15)}{4 \times 45{,}000}} + \cdot 5.$$

This will be found to give practically the same results as Rule 44a and Table XXXVI., and may be used in their place for cylinder covers, if preferred.

These covers should be stiffened by radial ribs which may have a thickness of $\cdot 9 \times$ thickness of cover, and a depth at centre of $3 \cdot 5 \times$ thickness of cover,—the outer edge of rib being a curve of the parabolic type when the cover is flat.

It will often happen that a cover can be made more or less curved, or dished one way or the other, and this should always be done where possible,—as the whole cover may then be made lighter.

The number of ribs may be the same as for cast-iron pistons (see Table XLVII.).

When weight-shaft brackets, or slide-rod guides, are attached to these covers, they must, of course, be made proportionately stronger.

For rectangular doors consisting of a single flat plate of metal of thickness (t), stiffened by ribs, the greatest section of which is not less than $(3 \times t) \times (\cdot 9t)$, and which are spaced in accordance with Rule 48 (page 90), the formula is,—

Rule 54. $$t = \sqrt{\frac{K \times b^2 \times (p+15)}{c}} + \cdot 5 \quad . \quad . \quad . \quad (b)$$

where t = thickness in inches.

b = breadth within flanges of valve-boxes, in inches.

$$K = \frac{\text{Length}^4}{\text{Length}^4 + \text{breadth}^4}$$

p = max. pressure in lbs. per sq. inch to which door is subject.

c = 70,000 for single doors.

c = 450,000 for double doors (see next paragraph).

When the door is rectangular and of the double type—consisting of two equal thicknesses of metal tied together and locally stiffened by

ribs, in conformity with Rule 48—the over-all thickness of the door at centre should not be less than $\frac{\text{Shortest span}}{5}$, and the outer surface may be curved in both directions, like a "hog-back" girder. When these conditions are complied with, the formula (b), given above, will give the thickness of either the outer or inner metal if the value $c=450,000$ be employed.

Large doors of this type, when fitted to Naval engines, are commonly made in cast steel. When this material is used the over-all thickness of door at centre may be $\frac{\text{Shortest span}}{6{\cdot}5}$, and the following variant of the above formula (b) may be used to determine thickness of metal:—

Rule 54a. $$t=\sqrt{\frac{K\times b^2\times(v+15)}{250,000}}+{\cdot}25.$$

A convenient method of saving some space and weight, in connection with doors of this type, is to slightly increase the thickness of the inner metal, and make sufficient openings at the ends, to allow of the door itself being utilised as a passage for the steam from one end of the valve-box to the other.

JOINTS OF CYLINDER COVERS, ETC.

As the widths and thicknesses of flanges should vary directly as the diameter of the stud employed, it is best when designing a joint to fix this diameter first.

In fixing the diameter of stud the three following conditions must be satisfied:—

(1) $\frac{\text{Total load on cover}}{\text{Effective area of 1 stud} \times \text{No. of studs}}$ must not exceed value given in Table XLIII.

Rule 55.—(2) The diameter of stud should not exceed ·8 × thickness of flange; and the thickness of flange should not exceed 1·5 × thickness of metal to which it is attached.

Rule 56.—(3) The pitch of the studs should not be less than diameter × 3·5 (to give room for spanner).

When no water test is required a pitch of 6 diameters is often used for L.P. doors, &c.

The diameter is usually arrived at by a rough process of trial and error, as studs and bolts must be of standard size.

Rule 57.—The width of joint may be from diameter × 2·3 to 3, but—unless weight is of the greatest importance—it should not be less than diameter × 2·8.

The thickness of cover flange may be from 1 to 1·3 × diameter.

The limit which the pitch should in no case exceed is given by the rule,—

Rule 58. Maximum pitch in inches = $\sqrt{\frac{t \times 100}{p}}$

where t is the thickness of cover or door flange in sixteenths of an inch, and p the pressure in pounds per square inch, on the cover.

For covers of cast steel the co-efficient in the above rule may be 120 in place of 100.

It is not necessary that studs should always carry the full load allowed by Table XLI., and it is sometimes convenient to load them much more lightly when it is desired to use the same size of stud for all the covers in connection with a set of cylinders.

Where bolts are used, as in the case of a steam joint between two cylinders, the thickness of the flanges should be about diameter of bolt × 1·25, and not exceed 1·5 × (thickness of metal to which flange is attached); the width of the joint is, of course, also increased by an amount equal to about ·6 of the thickness of that metal.

CYLINDER RELIEF OR ESCAPE VALVES.

Rule 59.—For the L.P. cylinder these should have a diameter of about one-fifteenth the diameter of the cylinder; for the H.P. cylinder, where priming water may have to be dealt with, the proportion may be one-eleventh; and for the M.P. cylinder, an intermediate proportion of say one-thirteenth.

In horizontal engines there should be at least one valve at each end of each cylinder—(the Admiralty requirement was two valves at each end of each cylinder)—and in vertical engines there should be one at each end of the H.P. cylinders, and one at the bottom of each of the others.

The following Table of sizes of Relief-valves is calculated to give about the above-named proportions, but the nearest half-inch is given in most cases—as it is neither usual nor necessary to make patterns for every quarter of an inch.

Table XXXVIII.—Cylinder Relief Valves.

Dia. of cylinder.	Dia. of Relief Valve in Ins.			Dia. of cylinder.	Dia. of Relief Valve in Ins.		
	H.P.	M.P.	L.P.		H.P.	M.P.	L.P.
15	1½	...	...	65	6	5	4½
20	2	1¾	...	70	...	5½	4½
25	2¼	2	1¾	75	...	6	5
30	2½	2¼	2	80	...	6	5½
35	3	2½	2¼	85	For valves over 5 ins. dia. fit two valves of equivalent area.		5½
40	3½	3	2½	90			6
45	4	3½	3	95			6½
50	4½	4	3½	100			6½
55	5	4½	3½	105			7
60	5½	4½	4	110			7½

The H.P. and L.P. columns are equally applicable to the cylinders of Compound, Triple, and Quadruple engines.

It is very desirable that all relief-valves should be fitted with guard rings, or other suitable appliances, to prevent thoughtless screwing up of the springs whenever a slight leakage occurs.

In confined engine-rooms it is also desirable to have light pipes fitted to convey any water that may escape down to the crank-pits or bilge.

For methods of calculating strengths and proportions of springs see Board of Trade Rules for Springs.

RECEIVER SAFETY VALVES.

For small and medium sized cylinders (up to 60 inches diameter say), these may be of the same size and pattern as the cylinder relief-valves,—*i.e.*, L.P. receiver valve same as L.P. cylinder valve, and so on; but this rule makes the valves unnecessarily large for larger cylinders, and more appropriate sizes are given by the rule,—

Rule 60.

$$\text{Diameter of Receiver Safety Valve} = \frac{\sqrt{\text{Diameter of cylinder.}}}{2}$$

DRAIN VALVES, OR COCKS, FOR CYLINDERS, RECEIVERS, ETC.

These should be of the proportions given by the following Table :—

Table XXXIX.—Drain Valves.

Diameter of Cylinder or L.P. Cylinder, if Compound, Triple, &c.	Diameter of valve or cock.	Diameter of Cylinder or L.P. Cylinder, if Compound, Triple, &c.	Diameter of valve or cock.
Up to 10″	½″	46 to 60″	1¼″
11 to 15″	⅝″	61 to 75″	1½″
16 to 30″	¾″	76 to 90″	1¾″
31 to 45″	1″	Above 90″	2″

In Compound, Triple, and Quadruple engines, the valves or cocks in connection with the H.P. and M.P. cylinders, should be of the same size as those for L.P. cylinder.

All pipes from H.P. and M.P. drain valves should be led to hot-well or feed tank, and should be fitted with self-acting non-return valves to prevent water getting back into cylinders. The pipes from L.P. drain valves should be led to the condenser, and also be fitted with non-return valves.

The drainage from the steam jackets should be collected in suitable vessels (each fitted with a gauge glass), and then led away to the hot-well or feed tank. The pipes should be fitted with adjustable screw-down valves, placed in sight of the gauge glasses, and within easy reach of the attendants.

Reversing engine drains should be led to the condenser, and fitted with non-return valves; but the drainage from other auxiliary engines is better led into the bilges, as it always contains a large percentage of oil.

Each steam jacket must have its own separate adjustable drain-valve, but the cylinder and receiver drains may be, to a certain extent, grouped, or led into common pipes so as to reduce the number of valves on hot-well or feed tank, and condenser.

Drain cocks of suitable sizes should be fitted to every pipe, passage, or place where water can lodge, so that, when engines are cold, they may be quite free from water.

STARTING, PASS, OR AUXILIARY VALVES.

In small Paddle engines it is usual to arrange the slide-valves to be worked by hand, and starting valves are therefore unnecessary.

In larger low-pressure Paddle engines, where the valves are driven by double eccentrics and links, the cuts-off are so late that starting-valves are not required.

Compound engines, with cuts-off at about six-tenths of the stroke, would occasionally be unhandy without some means of admitting steam to one of the cylinders later than the main slide allows, and

starting valves should therefore be fitted to the L.P. cylinders so that steam may be admitted into either side of piston.

Triple engines, with three cranks, require only a small valve to admit steam to each receiver; the valves should be raised from their seats by means of levers acting on the spindles, and should be held shut by small spiral springs in addition to the pressure of the steam. The steam supply should be taken from the boiler side of the regulator value, and may be about one-fifth the diameter of the main steam pipe.

COLUMN FEET AND BOLTS.

Great care should be exercised in designing these feet, for through them the load due to the steam pressure *on the cover* is transmitted; as the load is always applied suddenly, very ample section of metal should be provided to sustain it.

Rule 61.—The area of section through these feet should be such that the stress does not exceed 600 lbs. per square inch.

The webs from the flanges of the feet should be well spread over the cylinder bottom and towards the sides, so as to distribute the strain.

Rule 62. The bolts connecting the cylinder to the columns or frames should be such that the stress on them does not exceed 4000 lbs. per square inch of area at the bottom of the thread, and when there are a large number, of comparatively small size, it should not exceed 3000 lbs. per square inch.

See also Table XLIII., page 103, for loads that bolts may carry. The maximum or initial pressure should be used in calculating these stresses.

The feet should always be formed so as to permit of bolts being used (not studs), and two at least, in each foot, should be a driving fit.

GENERAL REMARKS ON CYLINDERS.

Where weight is of great importance, as in Naval machinery, it is a common practice to shorten the cylinder, and form the port in the

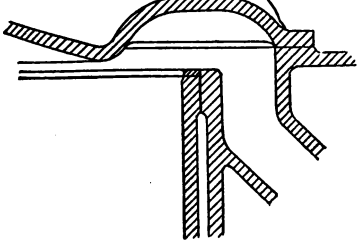


FIG. 9.

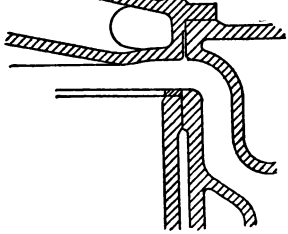


FIG. 9A.

cover, as shown in Fig. 9,—the more ordinary plan being shown by Fig. 9A.

Another method of saving weight, sometimes practised in connection with very light machinery, is to lead the exhaust steam through the relief-ring on the back of the L.P. valve, thus doing away with the exhaust port in the cylinder face, and shortening the slide valve and casing to that extent.

The corners of ports, both in false faces and in cylinder faces, should be well rounded, as the castings are very apt to crack if they are made quite square.

Steel screw-stays may be used with advantage to strengthen the various ports and passages in cylinder castings.

Where a false face is secured by the usual "cheese-headed" screws, a certain measure of relief may be given to the slide-valve, and the lubrication improved, by connecting the recesses by small grooves cut in the face.

Horizontal Cylinders.

Horizontal cylinder barrels should always be stiffened by rings, and when the valve-box is on top—two strong ribs running up from each foot to the flange of the valve-box door, to prevent deformation of those portions of the barrel lying between the foot brackets and the valve-box, each of a thickness of ·8 × thickness of barrel, and stand out 2 × thickness of barrel. Similar ribs should also be carried round under the bottom of the cylinder from foot to foot.

In bolting horizontal cylinders down to the seatings, fitted bolts should be used at the front ends only—sufficient clearance being allowed in the other bolt holes to permit the expansion of the cylinder.

Oscillating Cylinders.

In oscillating engines, weight is generally of great importance; the cylinder barrels are commonly made rather lighter than if the thickness were determined by the rules given (page 82). This reduction is rendered possible by the absence of any strains communicated from the framing to the cylinder, and by the great stiffness imparted to the barrel by the steam and exhaust belts, &c., and their accompanying ribs. The H.P. cylinders of compound oscillating engines should be of the thickness given by the following formula:—

Rule 63. Thickness of H.P. cylinder barrel $= \frac{D}{6750}(p+50) + \cdot 45$ inch, where D is diameter of cylinder in inches, and p boiler pressure.

The formula also gives the thickness of barrel for the L.P. cylinders of compound engines, and for the cylinders of simple engines working at 30 to 35 lbs. per square inch, if p be taken as 25. The above thicknesses are for cylinders without liners.

The trunnions should be all of the same diameter throughout, whether engines are simple or compound, and must, of course, be large enough to allow proper area for exhaust. This plan allows of an

annular air space between the steam pipe and the outer or working surface of the trunnion, and so gives a comparatively cool bearing.

The area of section where the trunnion joins the cylinder barrel should be such that the value of the expression,—

Rule 64.

$$\frac{\text{Max. effective pressure} \times \text{area of piston} + \text{weight of cylinder in lbs.}}{2 \times \text{area of section}}$$

does not exceed 500 ; *i.e.* the shearing stress should not exceed 500 lbs. per square inch.

A length of trunnion of about ·4 of the diameter will usually give a satisfactory amount of surface, but the area should be such that the pressure per square inch (excluding pressure due to weight of cylinder) does not exceed 350 lbs. A safe proportion, which should always be obtained where possible, is given by,—

Rule 65.

$$\frac{\text{Max. effective pressure} \times \text{area of piston}}{2 \times \text{diameter of trunnion} \times \text{length of same}} = 300.$$

A very strong attachment of trunnion is obtained by making the outer cylinder of barrel form, and arranging the steam and exhaust belts between it and the inner cylinder or liner.

The valve faces should not be at right angles to the line joining valve-spindle and piston-rod centres, but angled so that the side next the steam entrance is nearer to the cylinder barrel than the opposite or exhaust side; this will bring the centre of valve-spindle as close in to the cylinder as possible, whilst allowing free way for the exhaust,

The receivers of compound oscillating engines are commonly formed around the H.P. cylinder, between the inner cylinder or liner and the outer shell to which the trunnions are attached. This method has the accompanying advantage of making the outer cylinders more nearly of a size.

CLEARANCE OF PISTON.

The following Table shows the axial clearances required in cylinders of the various diameters mentioned.

Table XL.—Piston Clearances (axial).

Diameter of Cylinder.	Clearances.		Diameter of Cylinder.	Clearances.	
	Crank end.	Cover end.		Crank end.	Cover end.
Up to 14″	3/8″	1/4″	61 to 80″	7/8″	5/8″
15 to 20″	1/2″	3/8″	81 to 100″	15/16″	11/16″
21 to 40″	5/8″	1/2″	Above 100″	1″	3/4″
41 to 60″	3/4″	9/16″	...	...	...

STUFFING BOXES, ETC.

Table XLI.—Stuffing-boxes for elastic packing. (Fig. 10.)

A Diameter of Rod.	B Width of packing space.	C Diameter of Box. 1·25×A+·55	D_1 50 lbs. Depth of packing. ·9×A+·75	D_2 100 lbs. Depth of packing. 1·15×A+·75	D_3 150 lbs. Depth of packing. 1·4×A+·75	E Overall depth of neck-ring. ·45A+·35	F Thickn's of bushes, gland, and neck. ·045×A+·18	Number and diameter of studs.
¾"	⅜"	1½"	1⅜"	1⅝"	1¾"	¾"	3/16"	2- ⅜"
⅞	⅜	1⅝	1½	1¾	2	¾	3/16	2- ⅜
1	7/16	1⅞	1⅝	2	2¼	⅞	¼	2- ½
1⅛	7/16	2	1¾	2⅛	2⅜	⅞	¼	2- ½
1¼	7/16	2⅛	1⅞	2¼	2½	⅞	¼	2- ½
1⅜	7/16	2¼	2	2⅜	2⅝	1	¼	2- ⅝
1½	½	2⅜	2⅛	2½	2⅞	1	¼	2- ⅝
1¾	½	2¾	2¼	2¾	3¼	1⅛	¼	2- ¾
2	9/16	3⅛	2½	3⅛	3½	1¼	¼	2- ¾
2¼	9/16	3⅜	2¾	3½	3⅞	1⅜	5/16	2- ⅞
2½	⅝	3¾	3	3¾	4¼	1½	5/16	2- ⅞ 3- ¾
2¾	⅝	4	3¼	4	4⅝	1⅝	5/16	2-1 3- ¾
3	11/16	4⅜	3½	4¼	5	1¾	5/16	2-1 3- ⅞
3¼	11/16	4⅝	3¾	4¾	5⅜	1⅞	⅜	2-1⅛ 3- ⅞
3½	¾	5	4	5	5⅝	2	⅜	2-1⅛ 3-1
3¾	¾	5¼	4¼	5¼	6	2⅛	⅜	2-1¼ 3-1
4	13/16	5⅝	4½	5½	6⅜	2⅛	⅜	2-1¼ 3-1
4½	⅞	6¼	4¾	6	7	2⅜	7/16	2-1⅜ 3-1⅛
5	15/16	6⅞	5	6½	7¾	2⅝	7/16	2-1⅜ 3-1⅛
5½	1	7½	5⅝	7	8½	2⅞	7/16	4-1⅛
6	1⅛	8¼	6¼	7⅝	9¼	3	7/16	4-1⅛
6½	1⅛	8¾	6⅝	8⅛	9⅞	3¼	½	4-1⅛
7	1¼	9½	7	8¾	10½	3½	½	4-1¼
7½	1¼	10	7½	9⅜	11¼	3¾	½	4-1¼
8	1⅜	10¾	8	10	12	4	9/16	4-1¼
8½	1⅜	11¼	8½	10½	12¾	4¼	9/16	4-1⅜
9	1½	12	8⅞	11	13⅜	4⅜	9/16	4-1⅜
9½	1½	12½	9⅜	11⅝	14⅛	4⅝	⅝	4-1⅜
10	1½	13	9¾	12¼	14¾	4⅞	⅝	4-1⅜

In all cases, the amount that the gland can enter (G) should be one-half the depth of the stuffing box (D_1, D_2, or D_3).

The width of packing space (B) given in above Table assumes that all glands are of gun-metal, as is usual in naval practice; if it is desired to use cast-iron glands bushed with gun-metal the width of space must be increased ⅛ inch in each case.

A	5/16"	bush for a cast-iron gland may be used up to		1¼"	dia. of rod.
	¼"	,,	,,	2¾"	,,
	5/16"	,,	,,	3¾"	,,
	⅜"	,,	,,	6"	,,
	7/16"	,,	,,	7½"	,,
	½"	,,	,,	9"	,,
	9/16"	,,	,,	10½"	,,

Where two numbers and sizes of studs are given, the larger studs are intended for use with pinion nuts and toothed rings.

The sizes of studs for rods of 5½ inches diameter and upwards are fixed on the assumption that pinion nuts and toothed rings will always be used.

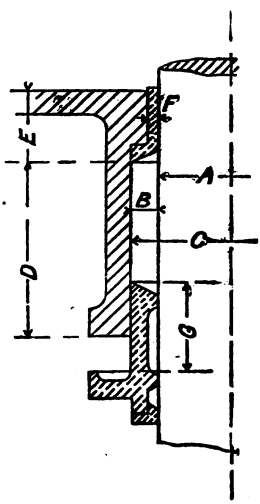


FIG. 10.

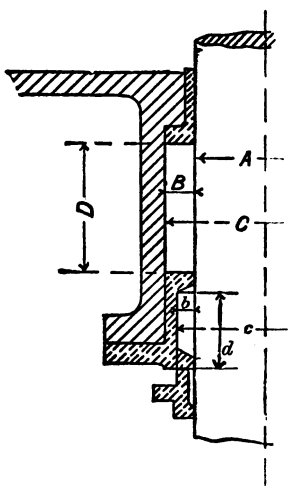


FIG. 11.

Table XLII.—Stuffing-boxes for Metallic Packing. (Fig. 11.)

A. Diameter of rod.	B. Width of packing space.	C. Diameter of box. $1{\cdot}2A+1{\cdot}25$	D. Depth of metallic packing. $2\sqrt{A}+{\cdot}5$.	*b.* Width of packing space.	*a.* Diameter of box.	*d.* Depth of elastic packing. $\frac{D}{8}+1{\cdot}15$.
3	1	5	4	3/4	4 1/2	2 1/8
3 1/4	1	5 1/4	4 1/8	3/4	4 3/4	2 1/8
3 1/2	1	5 1/2	4 1/4	3/4	5	2 1/4
3 3/4	1	5 3/4	4 3/8	3/4	5 1/4	2 1/4
4	1	6	4 1/2	3/4	5 1/2	2 3/8
4 1/4	1	6 1/4	4 5/8	3/4	5 3/4	2 1/2
4 1/2	1 1/8	6 3/4	4 3/4	3/4	6	2 5/8
5	1 1/8	7 1/4	5	3/4	6 1/2	2 3/4
5 1/2	1 1/8	7 3/4	5 1/4	3/4	7	2 7/8
6	1 1/4	8 1/2	5 1/2	3/4	7 1/2	3
6 1/2	1 1/4	9	5 5/8	3/4	8	3 1/8
7	1 1/4	9 1/2	5 3/4	3/4	8 1/2	3 1/4
7 1/2	1 3/8	10 1/4	6	7/8	9 1/4	3 1/2
8	1 3/8	10 3/4	6 1/4	7/8	9 3/4	3 5/8
8 1/2	1 3/8	11 1/4	6 3/8	7/8	10 1/4	3 3/4
9	1 1/2	12	6 1/2	1	11	3 7/8
9 1/2	1 1/2	12 1/2	6 3/4	1	11 1/2	4
10	1 1/2	13	7	1	12	4 1/8

For the sake of uniformity the H.P., M.P., and L.P. stuffing-boxes of triple engines are commonly made all of the same depth.

STRENGTH, &c., OF STUDS AND BOLTS.

As small bolts and studs are subject to severe wrenching, especially by careless hands, the nominal tensile working stress on them must be much less than with larger ones and may increase as the bolts are bigger.

The Table is based on the relation,—and the loads given should not be exceeded.

Working stress per sq. in. = (Area at bottom of thread)$^{\frac{1}{12}}$ × C;

where C = 5000 for iron or mild steel, and 1000 for strong bronzes. For iron or steel bolts above 2 inches in diameter, and gun-metal or bronze ones above 3½ inches diameter, the moment of the twisting stress is so small, proportionately, that it may be neglected.

Table XLIII.—Strengths of Studs and Bolts.

Outside Dia. of stud or bolt.	Iron or Mild steel.		Strong Bronze.	
	Working stress in lbs. per square inch.	Working load of 1 bolt in lbs.	Working stress in lbs. per square inch.	Working load of 1 bolt in lbs.
½	2000	250	400	50
⅝	2500	500	500	100
¾	3000	900	600	180
⅞	3400	1,450	680	290
1	3900	2,150	780	430
1⅛	4300	3,000	860	600
1¼	4700	4,200	940	840
1⅜	5100	5,400	1020	1,080
1½	5500	7,100	1100	1,420
1⅝	5800	8,500	1160	1,700
1¾	6300	11,000	1260	2,200
1⅞	6600	13,100	1320	2,620
2	7000	16,100	1400	3,220
2¼	7000	20,400	1560	4,560
2½	7000	26,100	1730	6,450
2¾	7000	31,200	1860	8,300
3	7000	38,100	2030	11,000
3¼	7000	44,800	2170	13,900
3½	7000	53,000	2350	17,800
3¾	7000	60,500	2500	21,600
4	7000	70,100	2500	25,000
4¼	7000	79,500	2500	28,400
4½	7000	90,300	2500	32,200
4¾	7000	100,800	2500	36,000
5	7000	113,000	2500	40,300
5¼	7000	124,600	2500	44,500
5½	7000	138,000	2500	49,200

Studs and bolts may be loaded to the figures given in the above table whether the load is "dead" (as in the case of a joint), or "live" (as in the case of a connecting-rod bolt), as in the latter case mild steel will always be used, and the shearing stress due to tightening up is practically absent.

Mild steel studs and bolts should always be fitted with iron nuts, as steel ones have a tendency to "seize"; for the same reason soft bronze studs, &c., should always have iron or hard bronze nuts if possible.

Gun-metal, and the various bronzes are unsatisfactory materials for small studs and bolts, not because of any lack of tensile strength,—which is often high,—but because of their very low resistance to shearing.

When iron or steel studs are used in connection with gun-metal steam or water valves, &c., they must not be allowed to penetrate into the steam or water space, or they will rapidly corrode and come loose.

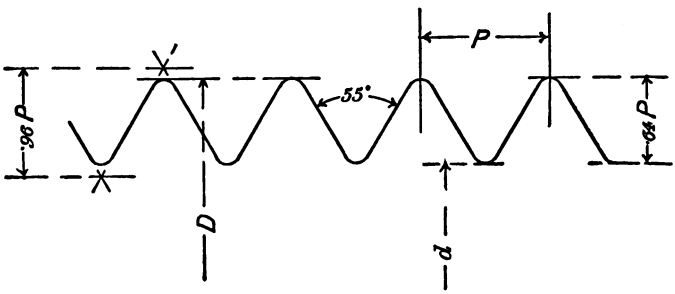
The part of a stud that is screwed into the work should be :—

Rule 66. Not less than $1\frac{1}{4}$ diameters long when screwed into cast-iron, and $1\frac{1}{2}$ diameters when not inconvenient.

Not less than 1 diameter long when screwed into gun-metal, wrought-iron, or cast-steel.

The general dimensions, numbers of threads, &c., for bolts and nuts given in Table XLIV. are in accordance with the Whitworth Standard, and the sizes over flats and angles are the nearest sixteenth to the same.

The proportions of the Whitworth Standard thread are shown by Fig. 12.



$$d = D - 1{\cdot}28P$$

FIG. 12.

Table XLIV.—Dimensions of Nuts, Bolts, &c.

Dia. of Bolt.	Head and nut over flats.	Head and nut over angles.	Height of nut.	Height of head for screws and bolts.	No. of threads per inch.	Area at bottom of thread in sq. inches.	Thick. of check-nut.	Size of split-pin L. S. G.
1/4	1/2	5/8	1/4	3/16	20	·027	3/16	No. 14
5/16	11/16	13/16	5/16	5/16	16	·068	1/4	,, 13
1/2	15/16	1 1/16	1/2	7/16	12	·121	3/8	,, 12
5/8	1 1/8	1 1/4	5/8	9/16	11	·203	7/16	,, 11
3/4	1 5/16	1 1/2	3/4	11/16	10	·303	9/16	,, 10
7/8	1 1/2	1 11/16	7/8	3/4	9	·421	5/8	,, 9
1	1 11/16	1 15/16	1	7/8	8	·554	3/4	,, 8
1 1/8	1 7/8	2 1/8	1 1/8	1	7	·697	13/16	,, 7
1 1/4	2 1/16	2 3/8	1 1/4	1 1/16	7	·894	15/16	,, 6
1 3/8	2 5/16	2 9/16	1 3/8	1 3/16	6	1·059	1 1/16	,, 5
1 1/2	2 7/16	2 13/16	1 1/2	1 5/16	6	1·300	1 1/8	,, 4
1 5/8	2 9/16	3	1 5/8	1 7/16	5	1·471	1 1/4	,, 3
1 3/4	2 3/4	3 3/16	1 3/4	1 9/16	5	1·752	1 5/16	,, 2
1 7/8	3	3 1/2	1 7/8	1 5/8	4 1/2	1·986	1 3/8	,, 1
2	3 1/8	3 5/8	2	1 3/4	4 1/2	2·311	1 1/2	,, 1
2 1/4	3 9/16	4 1/16	2 1/4	2	4	2·925	...	5/16
2 1/2	3 7/8	4 1/2	2 1/2	2 5/16	4	3·732	...	5/16
2 3/4	4 5/16	4 13/16	2 3/4	2 7/16	3 1/2	4·463	...	3/8
3	4 1/2	5 1/4	3	2 5/8	3 1/2	5·449	...	3/8
3 1/4	4 7/8	5 5/8	3 1/4	2 13/16	3 1/4	6·406	...	3/8
3 1/2	5 5/16	6	3 1/2	3 1/16	3 1/4	7·572	...	7/16
3 3/4	5 9/16	6 3/8	3 3/4	3 1/4	3	8·656	...	7/16
4	5 15/16	6 7/8	4	3 1/2	3	10·026	...	1/2
4 1/4	6 3/8	7 3/8	4 1/4	3 3/4	2 7/8	11·370	...	1/2
4 1/2	6 13/16	7 7/8	4 1/2	3 15/16	2 7/8	12·913	...	9/16
4 3/4	7 1/4	8 1/16	4 3/4	4 1/8	2 3/4	14·413	...	9/16
5	7 13/16	9	5	4 3/8	2 3/4	16·145	...	5/8

The heads of all ordinary screws and bolts should be made hexagonal, and of the same size over flats as the corresponding nuts.

All check-nuts should be chamfered on both sides.

Set-screws that are frequently handled, such as those in the lock-rings on the various large nuts, should have *square* heads, and should either be of hard steel or should have their heads properly case-hardened.

Wherever a square nut is used about the engines in place of a hexagon, care should be taken that it is made of the same size as some nut over flats, as otherwise much trouble and annoyance are often caused.

PISTONS.

For the case of a circular flat plate, supported at centre, and uniformly loaded, the formula may be written,—

$$t = C \times D\sqrt{p};$$

where t = thickness, and D diameter,—both in inches; p = effective pressure (or greatest difference of pressures on the two sides); and C a co-efficient.

Cast-steel Pistons.—These are made of a single thickness of metal, and should be coned, or "dished," to get the necessary rigidity. The three pistons of an ordinary triple engine may all be made of the same total depth, which should be such that the slope of the face of L.P. piston next crankshaft is about 1 in 3; this will give a perfectly rigid piston, and will also give room for a properly proportioned piston-rod stuffing-box.

The thickness near boss, for a piston of this type, is given very nearly by the above formula, when C = ·0046; that is,—

Rule 67. $t = \cdot0046\ D\sqrt{p}$ . . . . (*a*).

For practical reasons, however, the addition of a small constant quantity, and other slight modifications, are necessary, and the formulæ for the pistons of triple engines working at about 180 lbs. pressure, and of compound engines working at about 100 lbs. pressure (since p will have practically the same values in both cases) will be,—

For H.P. pistons, $t = \frac{D}{23} + \cdot24$ (value of p about 90);

For M.P. pistons, $t = \frac{D}{34} + \cdot40$ (value of p about 60);

L.P. do., $t = \frac{D}{50} + \cdot48$ (value of p about 30).

For triple engines working at 250 lbs. pressure use Rule 67, or equivalent formulæ with constant additions similar to the above, and put values of p at 150, 80, and 30.

The thicknesses near the edge, or rim, should be,—

Rule 68. For H.P. pistons, ·65 × thickness near boss;
M.P. do., ·60 × do. do.
L.P. do., ·55 × do. do.

For the pistons of quadruple, or other engines, similar modifications of the general formula (a) above may be employed.*

Table XLV. is calculated by means of the above formulæ, and may be used for compound engines by simply omitting the M.P. columns.

Forged-steel pistons for the lighter machinery of Torpedo boats and Destroyers give perfectly satisfactory results, when made of thickness, near boss, given by,—

Rule 69. $t = \cdot0035\ D\sqrt{p} + \cdot2,$

where the symbols have the same values as above.

The slope of the underside of L.P. piston should not be less than 1 in 6·5. The thickness near edge should be ·6 of that near boss; and the depth of rim may be $\frac{\text{L.P. Diameter}}{12}$. They are sometimes fitted with Ramsbottom rings of cast-iron or hard bronze, and sometimes with ordinary cast-iron packing rings.

They are, as a matter of course, machined over on both sides.

Table XLVI. is calculated by means of the above formula for boiler pressures of 220 to 250 lbs.

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* *Note*—See *Trans. Inst. C.E.*, vol. cxxvii., pp. 248 *et seq.*

Table XLV.—Cast-Steel Pistons (for 180 lbs. Working Pressure).

Diam. of piston in inches.	H.P. Thick. in inches.				M.P. Thick. in inches.				L.P. Thick. in inches.			
	Near boss.		Near edge.		Near boss.		Near edge.		Near boss.		Near edge.	
15	·89	15/16	·58	5/8	·84	7/8	·50	1/2	·78	13/16	·43	7/16
20	1·11	1 1/8	·72	3/4	·99	1	·59	5/8	·88	15/16	·48	1/2
25	1·33	1 3/8	·86	7/8	1·14	1 3/16	·68	11/16	·98	1	·54	9/16
30	1·54	1 9/16	1·00	1	1·28	1 5/16	·77	13/16	1·08	1 1/16	·59	5/8
35	1·76	1 3/4	1·14	1 3/16	1·43	1 7/16	·86	7/8	1·18	1 3/16	·65	11/16
40	1·98	2	1·29	1 5/16	1·57	1 9/16	·94	15/16	1·28	1 5/16	·70	3/4
45	2·20	2 1/4	1·43	1 7/16	1·72	1 3/4	1·03	1 1/16	1·38	1 3/8	·76	13/16
50	2·41	2 7/16	1·57	1 9/16	1·87	1 7/8	1·12	1 1/8	1·48	1 1/2	·81	13/16
55	2·63	2 5/8	1·71	1 3/4	2·02	2 1/16	1·21	1 1/4	1·58	1 5/8	·87	7/8
60	2·85	2 7/8	1·85	1 7/8	2·16	2 3/16	1·29	1 5/16	1·68	1 11/16	·92	15/16
65	...	...	...	...	2·31	2 5/16	1·38	1 3/8	1·78	1 13/16	·98	1
70	...	...	...	...	2·46	2 1/2	1·47	1 1/2	1·88	1 7/8	1·03	1 1/16
75	...	...	...	...	2·61	2 5/8	1·56	1 9/16	1·98	2	1·09	1 1/8
80	...	...	...	...	2·75	2 3/4	1·65	1 11/16	2·08	2 1/16	1·14	1 3/16
85	...	...	...	...	...	...	...	...	2·18	2 3/16	1·20	1 1/4
90	...	...	...	...	...	...	...	...	2·28	2 5/16	1·25	1 1/4
95	...	...	...	...	...	...	...	...	2·38	2 3/8	1·31	1 5/16
100	...	...	...	...	...	...	...	...	2·48	2 1/2	1·36	1 3/8
105	...	...	...	...	...	...	...	...	2·58	2 9/16	1·42	1 7/16
110	...	...	...	...	...	...	...	...	2·68	2 11/16	1·47	1 1/2

For solid cast-iron pistons of moderate size multiply edge dimensions by 1·2, and near boss by 1·7 when of good material.

Table XLVI.—Forged Steel Pistons of Torpedo Boats and Destroyers (220 to 250 lbs.).

Diameter of piston in inches.	H.P. Thick. in inches.				M.P. Thick. in inches.				L.P. Thick. in inches.			
	Near boss.		Near edge.		Near boss.		Near edge.		Near boss.		Near edge.	
10	·63	5/8	·38	3/8	...	...	...	...	...	...	...	...
15	·84	1 3/16	·50	1/2	·67	1 1/16	·40	3/8	...	...	...	...
20	1·05	1 1/16	·63	5/8	·83	1 3/16	·50	1/2	·58	9/16	·35	3/8
25	1·27	1 1/4	·76	3/4	·98	1	·59	9/16	·67	1 1/16	·40	3/8
30	1·48	1 1/2	·89	7/8	1·14	1 1/8	·68	1 1/16	·77	3/4	·46	7/16
35	...	...	...	...	1·29	1 1/4	·77	3/4	·87	7/8	·52	1/2
40	...	...	...	...	1·45	1 7/16	·87	7/8	·96	1 5/16	·58	9/16
45	...	...	...	...	...	...	...	...	1·06	1 1/16	·64	5/8
50	...	...	...	...	...	...	...	...	1·15	1 1/8	·69	1 1/16
55	...	...	...	...	...	...	...	...	1·25	1 1/4	·75	3/4

Cast-iron Pistons.—These must be made of thoroughly sound and good material,—at least twice melted.

A convenient unit on which to base the proportions of cast-iron box (or hollow) pistons is the thickness requisite, in a circular plate of cast-iron, to carry safely a given pressure per square inch when supported at centre and loaded uniformly.

Using again the general formula given above for such a case, viz.:—

$$t = C \times D\sqrt{p}$$

and inserting a value of c that will limit the stress on the material to 3000 lbs. per square inch it becomes,—

$$t = \cdot 008\ D\sqrt{p}$$

If the necessary constant quantity be then added, the units (x) for pistons of triple engines working at about 160 lbs., and of compound engines working at about 100 lbs., will be given by,—

Rule 70. For H.P. pistons, $x = \cdot 008\ D\sqrt{p} + \cdot 3$

M.P. ,, $x = \cdot 008\ D\sqrt{p} + \cdot 45$

L.P. ,, $x = \cdot 008\ D\sqrt{p} + \cdot 6$

Other proportions are then as follows:—

Rule 71. Depths at centre, ... $3 \cdot 2 \times x$.

,, 71a. Thickness of metal of faces { near boss $\cdot 33 \times (x + \cdot 5)$. near edge $\cdot 3 \times (x + \cdot 5)$.

,, 71b. Thickness of webs, ... $\cdot 32 \times (x + \cdot 5)$.

,, 71c. Thickness of boss round rod, ... $\cdot 7 \times x$.

,, 71d. Depth of packing-ring, ... $\cdot 75 \times (\text{L.P.}\ x + 1 \cdot 5)$.

,, 71e. Thickness do. ... ... $\cdot 22 \times (\text{L.P.}\ x + 1 \cdot 5)$.

,, 71f. Diameter of junk-ring bolts, ... $(\cdot 2 \times \text{L.P.}\ x) + \cdot 5$.

Table XLVII.—Proportions

Diameter of piston in inches.	Values of x. $= \cdot 008 D\sqrt{p} + \begin{cases} \cdot 3 \text{ H.P.} \\ \cdot 45 \text{ M.P.} \\ \cdot 6 \text{ L.P.} \end{cases}$			Depths at centre. $= 3 \cdot 2 \times x$.			Thickness of Metal $= \cdot 33 \times (x + \cdot 5)$ at centre and			
							H.P.		M.P.	
	H.P.	M.P.	L.P.	H.P.	M.P.	L.P.	Centre.	Edge.	Centre.	Edge.
15	1·44	1·37	1·25	4·60	4·38	4·00	·64	·58	·62	·56
20	1·82	1·68	1·46	5·82	5·37	4·67	·76	·69	·72	·65
25	2·20	1·99	1·68	7·04	6·37	5·36	·89	·81	·82	·75
30	2·58	2·30	1·89	8·25	7·35	6·05	1·01	·92	·92	·84
35	2·96	2·61	2·11	9·47	8·33	6·75	1·14	1·04	1·02	·93
40	3·34	2·91	2·33	10·69	9·31	7·45	1·27	1·15	1·12	1·02
45	3·72	3·22	2·55	11·91	10·30	8·14	1·40	1·27	1·23	1·11
50	4·10	3·53	2·76	13·12	11·29	8·83	1·52	1·38	1·33	1·20
55	4·48	3·84	2·98	14·34	12·27	9·52	1·65	1·49	1·43	1·30
60	4·86	4·14	3·19	15·55	13·25	10·21	1·77	1·60	1·53	1·39
65	...	4·45	3·41	...	14·24	10·90	...	...	1·63	1·48
70	...	4·76	3·62	...	15·23	11·58	...	...	1·73	1·57
75	...	5·07	3·84	...	16·22	12·27	...	...	1·84	1·67
80	...	5·38	4·05	...	17·21	12·96	...	...	1·94	1·76
85	...	...	4·27	...	...	13·66	...	...	...	...
90	...	...	4·49	...	...	14·37	...	...	...	...
95	...	...	4·71	...	...	15·06	...	...	...	...
100	...	...	4·92	...	...	15·74	...	...	...	...
105	...	...	5·14	...	...	16·43	...	...	...	...
110	...	...	5·35	...	...	17·12	...	...	...	...

of Cast-Iron Pistons.

of Faces. ·3×(x+·5) at edge.		Thickness of boss round rod ·7×x.		Number of Webs.			Packing Ring.		Dia. of Junk-ring bolts (L.P.x×2)+·5
L.P.				H.P.	M.P.	L.P.	Depth ·75 × (L.P.x+1·6).	Thick. ·22 × (L.P.x+1·6).	
Centre.	Edge.	H.P.	L.P.	D+20 9	D+20 10	D+20 11			
·58	·52	1·01	·87	4	...	...	2·06	·60	·75
·65	·59	1·27	1·02	5	4	4	2·22	·65	·79
·72	·66	1·54	1·17	5	5	5	2·38	·70	·84
·79	·72	1·81	1·32	6	5	5	2·54	·75	·88
·86	·79	2·07	1·48	6	6	5	2·71	·80	·92
·93	·85	2·34	1·63	7	6	6	2·87	·84	·96
1·01	·92	2·60	1·78	8	7	6	3·03	·89	1·01
1·08	·98	2·87	1·93	8	7	7	3·19	·94	1·05
1·15	1·05	3·13	2·08	9	8	7	3·36	·99	1·10
1·22	1·11	3·40	2·23	9	8	8	3·52	1·03	1·14
1·30	1·17	...	2·38	...	9	8	3·68	1·08	1·18
1·37	1·23	...	2·53	...	9	9	3·84	1·13	1·22
1·44	1·30	...	2·68	...	10	9	4·00	1·18	1·27
1·50	1·36	...	2·83	...	10	9	4·16	1·22	1·31
1·58	1·43	...	2·99	...	...	10	4·33	1·27	1·36
1·66	1·50	...	3·14	...	...	10	4·49	1·32	1·40
1·73	1·56	...	3·29	...	...	11	4·65	1·37	1·44
1·80	1·62	...	3·44	...	...	11	4·81	1·41	1·48
1·87	1·69	...	3·59	...	...	12	4·97	1·46	1·53
1·93	1·75	...	3·74	...	...	12	5·13	1·51	1·57

Rule 72. Pitch of junk-ring bolts, { H.P.—7½ diameters. M.P.—8½ do. L.P.—10 do.

Rule 73. Spring pressures per square inch of packing-ring in contact with cylinder, ... { H.P.—3½ lbs. M.P.—3 lbs. L.P.—2½ lbs.

Rule 74. Number of webs in piston, ... { H.P.—$\frac{D+20}{9}$. M.P.—$\frac{D+20}{10}$. L.P.—$\frac{D+20}{11}$.

This spacing of webs will ensure that Rule 48, page 90, is not violated. In calculating *local* strength of piston face, care must be taken to use the gauge pressure acting on it, and not the effective pressure on piston as in above rules. A good method of increasing local strength, and at the same time assisting the moulder, is to cast a solid bar as a tie at the centre of gravity of each sector of the piston; the diameter of these ties may be $\cdot 6 \times L.P.x$ All holes provided for the removal of core material should be strengthened by an internal ring, unless through the circular rim.

Pistons of all Compound engines are usually, for practical reasons, made all of the same depth (except in the case of tandem engines); but, if a piston is made of less depth than that given in Table XLVII., the metal of the faces must be increased in proportion.

The proportions given for packing-rings and junk-ring bolts are equally applicable to cast-steel pistons.

All junk-ring bolts (or nuts, if studs are used) should be locked by means of a light wrought-iron ring, secured to junk-ring by studs having square bodies, which stand up through guard-ring, and through which stout split-pins are fitted: ⅝-inch studs and No. 1 L.S.G. split-pins are commonly used for large pistons.

When the ordinary packing-ring is used an excellent piston is produced by placing behind it,—one between every pair of junk-ring bolts,—a number of short spiral springs acting radially, and adjusted to give the pressures named above. For the pistons of auxiliary engines the Cameron spring (a corrugated steel ribbon) is most useful, and is also often employed for L.P. pistons of large size. The old-fashioned coach spring should never be employed.

In horizontal engines solid packings should be used in lieu of springs for the lower one-third circumference of piston; diagonal and oscillating pistons are also better for some solid packings.

Table XLVII. is calculated by means of the above formulæ, and shows at a glance the various proportions of H.P., M.P., & L.P. pistons, from 15 inches to 110 inches diameter, for Triple engines working at about 160 lbs., and Compound engines working at about 100 lbs. pressure.

When using Table XLVII. for Compound engines simply omit M.P. columns. The dimensions are given in inches and decimals only, as the proportions for pistons of intermediate diameter are more readily seen with this notation.

PISTON-RODS.

These should be made of steel of 38 tons tensile; softer steels do not wear satisfactorily. (**Rule 75**) The maximum permissible stress per square inch at bottom of thread should not exceed one-eighth the ultimate tensile strength of the steel employed.

For small rods (under 2 inches diameter in the body) the following should not be exceeded, viz. :—

Working Stress per sq. in. = diam. in ins. at bottom thread × C.

For high tensile steel, C = 4500. For ordinary mild steel (28/32 tons tensile), C = 3500.

Table XLVIII.—Strengths of Piston-Rods.

Diameter of piston-rod screw in inches.	38-Ton Steel.		Mild Steel ($^{28}/_{32}$ Tons Tensile).	
	Working stress in lbs. per sq. inch.	Effective strength of rod in lbs.	Working stress in lbs. per sq. inch.	Effective strength of rod in lbs.
⅞	3,700	1,560	2800	1,180
1	4,100	2,270	3100	1,720
1⅛	4,400	3,070	3300	2,270
1¼	4,800	4,290	3600	3,220
1⅜	5,100	5,400	3900	4,130
1½	5,400	7,020	4100	5,330
1⅝	5,700	8,380	4300	6,320
1¾	6,000	10,500	4600	8,060
1⅞	6,300	12,500	4800	9,530
2	6,600	15,200	5000	11,500
2¼	7,000	20,400	5300	15,500
2½	7,400	27,600	5600	20,900
2¾	7,800	34,800	5900	26,300
3	8,200	44,700	6100	33,200
3¼	8,600	55,000	6400	41,000
3½	9,000	68,000	6800	51,500
3¾	9,250	80,000	7200	62,300
4	9,500	95,200	7600	76,100
4½	10,000	130,000	8000	103,300
5 &c.	10,500	...	8400	...

A convenient form of the formula for the body of rod is then,—

Rule 76. Diameter of piston-rod $= \frac{\text{Diameter of cylinder}}{F} \times \sqrt{p}$,

where p = greatest effective pressure on the piston; F a co-efficient whose values are,—

Naval type engines,—direct-acting,	. . .	50
Mercantile,—ordinary stroke, direct-acting,	. .	45
,, long ,, ,,	. .	42
,, very long ,, ,,	. .	41
,, medium stroke, oscillating,	. .	40
,, long ,, ,,	. .	38

The stroke of Naval type engines rarely exceeds dia. of L.P. cylinder × ·6 and is usually between ·5 and ·6 × L.P. diameter.

"Ordinary" stroke, for the engines of a merchant steamer, is,—diameter of L.P. cylinder × ·60 to ·66; whilst ·7 to ·8 would be "long," and ·8 to 1 "very long."

A "medium" stroke for an oscillating engine is,—diameter of cylinder × ·95; or, if a compound engine,—diameter of L.P. cylinder × ·8.

Compound diagonal engines would come under the heading "long," since they are usually given a stroke of L.P. diameter × ·75, though the practice of different makers varies between ·7 and 1·3 × L.P. diameter.

Rule 76a. Area of section of piston-rod $= \frac{\text{Area of cylinder} \times p}{F^2}$

where F^2 will then be the stress in lbs. per square inch,—in each case the square of the numerical co-efficient given above, viz. :—

Naval engines, direct-acting,	. . .	2500
Mercantile, ordinary stroke, direct-acting,	. .	2000
,, long ,, ,,	. .	1800
,, very long ,, ,,	. .	1700
,, medium stroke, oscillating,	. .	1600
,, long ,, ,,	. .	1450

Attachment of Piston to Rod.—In small auxiliary engines it is a common practice simply to reduce the diameter of the piston-rod about ¼ inch, and fit the piston up against the shoulder; but above 15 inches diameter of piston it is better to leave only as much shoulder as may be desired for "trueing-up" the rod at some future time, and then give a taper of 1 in 4 on the diameters (**Rule 77**) (*i.e.* if taper is 4 inches long, the smaller diameter will be 1 inch less than the larger), and continue it until it dies away into the parallel part near screw, the diameter of which has been determined as directed above.

Where a piston-rod is fitted into a crosshead, the same shoulder, taper, &c., should be used as for a piston.

The depth of piston-rod nut need not exceed the diameter of the screw; and it may be recessed as far into the piston as desired, as the strength of the piston is not materially effected thereby.

The piston-rods of oscillating engines are commonly made with a sort of cylindrical "bolt-head," which is recessed into the lower face of the piston, whilst the nut is similarly recessed (to its full depth) into the upper face.

The cast-steel pistons of large Naval engines are sometimes made without any central boss, and attached to a circular flange (resembling a shaft coupling) on the piston-rod by means of a number of comparatively small bolts. This method, which was first used by the late Dr Kirk, has many good points, but the chief is perhaps the ease with which the connection can be made and broken again.

Rule 78. The diameter of this flange is about **diameter of piston-rod** $\times 2{\cdot}25$, and its thickness about **diameter of piston-rod** $\times {\cdot}33$. The attaching bolts (usually between $2\frac{1}{2}$ inches and $3\frac{1}{2}$ inches diameter) should be of mild steel, but should not carry more than 6000 lbs. per square inch.

PISTON-ROD GUIDES AND GUIDE BLOCKS.

Piston-rod guides should, wherever possible, be made of hard close-grained cast-iron, and the guide blocks, or crosshead shoes, should be made of the same material, since no combination of metals gives better results when in good working condition than cast-iron rubbing on cast-iron.

White metal is often used for the faces of the shoes (either fitted or cast into recesses in the cast-iron or gun-metal shoes), and gives results little or no better than those obtained with cast-iron on cast-iron, when the surfaces, etc., are *pari passu*, but with white metal there is no danger of abrasion, or from sparking in an enclosed engine.

Let r be the radius of crank and l the length of the connecting-rod; α the angle made by the rod with the axial line of cylinder. When P is the effective pressure on the piston; Q the pressure on guide;

$$\text{Then } Q = P \tan \alpha.$$

When the cut-off is later than half stroke, the maximum pressure will be when the crank is at right angles to the axial line.

$$\text{That is } Q = P \frac{r}{\sqrt{l^2 - r^2}},$$

as the connecting-rods in marine engines are generally twice the stroke, and $l = 4r$.

$$\text{Then, maximum pressure on guide } Q = \frac{P}{3{\cdot}87}.$$

The common method of fixing the sizes of crosshead shoes is to assume that the pistons, when at half stroke, have still the maximum effective pressures acting on them; to calculate the pressures on the guides from these loads in the manner described above; and then to

fix the areas of the shoes so that the pressure per square inch may not exceed 80 lbs. (**Rule 79.**) Pressure per square inch $=(1760 \div \sqrt{S+100})$ lbs., S being the speed of piston in feet per minute. Of course, where the indicator diagrams from similar engines are available, the actual effective pressures at half stroke may be ascertained and used, but the pressures per square inch may then be more than given by Rule 79. It may also be 30 to 50 per cent. more with efficient forced lubrication.

It is very desirable to keep the crossheads and guides of compound and triple engines all of the same dimensions, and to effect this, the *mean* of the piston loads may be used to calculate from. The surfaces of the astern shoes may be from ·5 to ·7 of those of the ahead ones.

For gudgeons, and bolts and caps, see under **Connecting-Rods.**

Where the gudgeon brasses are recessed into the piston-rod head, the inner or half-round brass may have a thickness,—

Rule 80. Thickness of inner half brass $= \frac{\text{Diam. of gudgeon}}{8} + \cdot 2,$

whilst for the outer or flat half the thickness may be,—

Rule 80a. Thickness of outer half brass $= \frac{\text{Diam. of gudgeon}}{7} + \cdot 2.$

See note with reference to white metal under "Connecting-rod gudgeons," page 118, also Table XLIX., page 121.

CONNECTING-RODS.

The length of the connecting-rod, measured from centre of gudgeon to centre of crank pin, should not be less than twice the stroke of the piston.

So far as tensile stress alone is concerned, a rod of the same diameter as the piston-rod at the bottom of the thread would suffice, but there are also to be considered compressive stress, bending stress due to the inertia of the rod itself, and bending stress due to the friction of the gudgeon and crank pin.

As marine connecting-rods are usually between 10 and 16 diameters (measured at mid-length) long, they must, when compressive stresses are under consideration, be treated as struts jointed at the ends.

Hodgkinson & Gordon's formula for the *breaking* strength of such a strut, of circular section, is,—

$$P = \frac{fs}{1 + 4a\frac{l^2}{d^2}}$$

where P is breaking load in lbs. ; l length from centre to centre, and d diameter, both in inches ; s area of section in square inches ; and f and a co-efficients, whose values for wrought-iron or mild steel are 36,000 and $\frac{1}{9000}$ respectively.

It should be noted that the ratio of greatest thrust on the connecting-rod is to the effective load on the piston is 1·034.

As the ratio of length to diameter rarely exceeds 16, however, it is probable that the following empirical rule (suggested by Grashof) will give more accurate results,—

Rule 81.

$$P = \frac{KAI}{\frac{Al^2}{c} + I}$$

where P is greatest safe load in lbs.; A sectional area of rod in sq. ins.; I moment of inertia (equal to $\cdot 05d^4$ for circular section); l length in inches from centre to centre; and K and c constants whose values are respectively 12,000 and 5000 for steel, and 10,000 and 5600 for wrought-iron. The maximum working load on a connecting-rod should not exceed ·75P, and is commonly about ·6P.

The bending stress, due to the inertia, may be found from the formula,—

Rule 82.

$$f = \cdot 137 \frac{v^2 l^2}{gRd}$$

where v is velocity of crank-pin in feet per second; l length of rod from centre to centre, and d diameter of rod, both in inches: R radius of crank in feet; and $g = 32$.

The value of f will be found to lie between 800 and 1800 lbs. per square inch for the various types of engine and numbers of revolutions met with in ordinary practice, and this value must be added to the compressive stress in estimating the total stress on the material of the rod.

In a three-crank engine, when one of the crank-pins heats, the bending stress set up in the connecting-rod by the continued action of all the pistons may be very great (if a "seize" could really occur the rod would be destroyed instantly), but its magnitude cannot be even approximately calculated, and the case is, therefore, one that can only be met by an empirical formula. The above formulæ may, therefore, be used as *checks*, but the following empirical rule will be more readily applied, and the results given by it will be found to agree very closely with good modern practice:—

Rule 83.

$$D = \frac{\sqrt{L \times K}}{4}$$

where D is diameter of rod at middle, and L length from centre to centre, both in inches; and

$K = \cdot 028\sqrt{\text{Effective load on piston in lbs.}}$—for Mercantile engines.

$K = \cdot 022\sqrt{\text{Effective load on piston in lbs.}}$—for Naval engines.

Connecting-rods are usually made tapered from the gudgeon end to the middle, and parallel from the middle to the crank end.

Rule 84.—D is the diameter of piston-rod by rule;
Diameter of gudgeon fixed in rod = $1 \cdot 125 \times D$;
Diameter of each gudgeon in crosshead = $0 \cdot 75 \times D$;
The length of gudgeon such that:—

Rule 85.—The loads on gudgeon bearings, from maximum effective pressures on pistons, should never exceed $(12,500 \div \sqrt{R+100})$ lbs. per square inch, R being the revolutions per minute.

Rule 85a.—The diameter of the gudgeon (when shrunk into the connecting-rod) may be $1 \cdot 25 \times$ Diameter of piston-rod, and the length is then given by,—

$$\text{Length of Gudgeon} = \frac{\text{Max. effective load on piston} \times \sqrt{R+100}}{\text{Diameter} \times 12,500}.$$

Connecting-Rod Gudgeons.—The rods of small engines may have the gudgeon shrunk into the jaws or sides of the double-eye, and working in brasses fitted into a recess in the piston-rod head, and secured by a cap and bolts; but larger engines should have the gudgeon shrunk into or formed with the piston-rod head or crosshead, and both jaws of the connecting-rod fitted with brasses, caps, and bolts, &c. The great advantages of the latter plan are,—the ease with which the brasses can be overhauled and adjusted, and the fact that it does not require the piston-rod crosshead to be forged solid with the rod. It also braces or "trusses" the joint between the two rods, and, in a measure, compensates for the absence of side guides, which experience has shown are better omitted; and, further, should unequal wear of the brasses take place, it occurs at a distance from the axis, and is therefore of less relative importance.

Where gudgeons cannot conveniently be made of hard steel (say 40 tons tensile), they must be case-hardened, and then carefully ground true. Simple mild steel is not suitable for gudgeons or other similar parts on which pressure is great, angular movement small, and direction of motion frequently reversed; and, similarly, hard gun-metal makes a better gudgeon bearing than any ordinary white metal.

Connecting-Rod Caps and Bolts.—The stresses on bolts of piston-rods and connecting-rods, per square inch area at bottom of thread, may be those given in Table XLIII., page 103, the bolts being in all cases of mild steel; but, where the load is carried by four bolts in place of two, the stress per square inch should be one-eighth less,—to allow for possible inequality in screwing up.

The bodies of the bolts should be turned down to the same diameter as the bottom of the thread, except where necessary for steadying the caps and brasses, and these remaining plain portions should be of slightly larger diameter than the screw thread,—say $\frac{1}{32}$ inch up to $2\frac{1}{2}$ inch diameter, and $\frac{1}{16}$ inch for larger sizes.

The nuts should be of wrought-iron or hard steel—as mild steel nuts

on steel bolts have a great tendency to "seize" and tear unless well greased.

The necessary section of cap is most readily determined by means of the ordinary formula for a beam supported at the ends, and uniformly loaded, viz. :—

Rule 86. $$w = \frac{8fz}{l^2}$$

where w is the working load per inch of length for any given value of f; l is the length in inches, from centre of bolt to centre of bolt; and z the modulus of the section,—equal to $\frac{\text{Breadth} \times \text{Depth}^2}{6}$ for a rectangular section. For wrought-iron, or mild steel, the values of f may be,—

Flat-backed brasses and caps,—load carried by one cap,				$f = 9{,}000.$
Half-round	,,	,,	,,	$f = 10{,}000.$
Flat-backed	,,	,,	two caps,	$f = 8{,}000.$
Half-round	,,	,,	,,	$f = 9{,}000.$

In estimating the breadths of caps, care should be taken to deduct the diameters, or breadths, of all oil holes.

The caps of connecting-rods of large mercantile engines are often made of tough cast-iron or cast-steel lined with white metal instead of half a loose shell and an ordinary cap. The shells of large rods are generally of tough cast-iron, which is really better than bronze for the carrier of white metal.

Connecting-Rod Brasses.—The over-all thickness of these (*i.e.* the thickness including any white metal) should be as follows :—

Rule 87. Flat-backed brasses for gudgeon end,—

$$\text{Thickness} = \frac{D}{7} + \cdot 2.$$

Rule 87a. Flat-backed brasses for crank end,—

$$\text{Thickness} = \frac{D}{8} + \cdot 2.$$

Rule 87b. Round brasses for gudgeon end,—

$$\text{Thickness} = \frac{D}{8} + \cdot 2.$$

Rule 87c. Round brasses for crank end,—

$$\text{Thickness} = \frac{D}{9} + \cdot 2.$$

Rule 87d.—Thickness of white metal $= \cdot 02D + \cdot 125.$

Rule 87e.—When no white metal is used at crank end the thicknesses may be $\begin{cases} \frac{D}{10} + \cdot 15 \text{ when flat backed.} \\ \frac{D}{11} + \cdot 12 \quad \text{,, round ,,} \end{cases}$

See Table XLIX., page 121, for proportions given by these rules.

Rule 88.—The width of the crank end of the connecting-rod should not be less than ·7 × length of crank-pin,—as otherwise the brasses will not be properly supported.

The proportions of the double-eye at gudgeon end (fig. 13) should be as follows:—

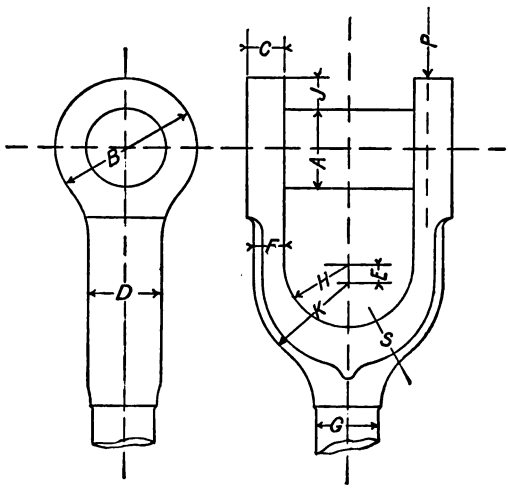


FIG. 13.

Rules 89 and 89a. $\frac{B}{A} = 1\ 82$, and $\frac{C}{A} = \cdot 475$.

Rules 89b and 89c. $\frac{D}{G} = 1 \cdot 2$, and $\frac{F}{G} = \cdot 45$.

The sectional area at D, or F, is then about ·62 of that at G; and area C × J about ·39 of that at G.

Rule 89d. Also $\frac{E}{H} = \cdot 275$.

Table XLIX.—Thickness of Brasses.

Diameter of gudgeon or pin.	Description of Bearing or Brasses.									
	Connecting rod gudgeon when in piston rod head.		Connecting rod gudgeon when on forked end of connect. rod.		Crank-pins.			Main Bearings.		
	Inner half. $\frac{D}{8}+\cdot2$	Outer half. $\frac{D}{7}+\cdot2$	Round brass's. $\frac{D}{8}+\cdot2$	Flat backed brass's. $\frac{D}{7}+\cdot2$	Round brasses. $\frac{D}{9}+\cdot2$	Flat backed brasses. $\frac{D}{8}+\cdot2$	White metal. $\cdot02D+\cdot125$	Round brasses. $\frac{D}{10}+\cdot25$	Flat backed brasses. $\frac{D}{9}+\cdot3$	White metal. $\cdot02D+\cdot125$
2	·45	·48	·45	·48	$\frac{D}{10}+\cdot12$ = ·32	$\frac{D}{9}+\cdot15$ = ·37	…	$\frac{D}{11}+\cdot12$ = ·30	$\frac{D}{10}+\cdot15$ = ·35	…
2½	·51	·56	·51	·56	·37	·43	…	·34	·40	…
3	·57	·63	·57	·63	·42	·48	…	·39	·45	…
3½	·64	·70	·64	·70	·47	·54	…	·45	·50	…
4	·70	·77	·70	·77	·64	·70	·20	·65	·74	·20
4½	·76	·84	·76	·84	·70	·76	·21	·70	·80	·21
5	·82	·91	·82	·91	·75	·82	·22	·75	·85	·22
5½	·89	·99	·89	·99	·81	·89	·23	·80	·91	·23
6	·95	1·06	·95	1·06	·87	·95	·24	·85	·97	·24
6½	1·01	1·13	1·01	1·13	·93	1·01	·25	·90	1·03	·25
7	1·07	1·20	1·07	1·20	·98	1·07	·26	·95	1·08	·26
7½	1·14	1·27	1·14	1·27	1·04	1·14	·27	1·00	1·14	·27
8	1·20	1·34	1·20	1·34	1·09	1·20	·28	1·05	1·19	·28
8½	…	…	1·26	1·41	1·15	1·26	·29	1·10	1·25	·29
9	…	…	1·32	1·48	1·20	1·32	·30	1·15	1·30	·30
9½	…	…	1·39	1·56	1·26	1·39	·31	1·20	1·36	·31
10	…	…	1·45	1·63	1·31	1·45	·32	1·25	1·41	·32
10½	…	..	1·51	1·70	1·37	1·51	·33	1·30	1·47	·33
11	…	…	1·57	1·77	1·42	1·57	·34	1·35	1·52	·34
12	…	…	1·70	1·91	1·53	1·70	·36	1·45	1·63	·36
13	…	…	1·82	2·06	1·64	1·82	·38	1·55	1·74	·38
14	…	…	…	…	1·75	1·95	·40	1·65	1·85	·40
15	…	…	…	…	1·87	2·08	·42	1·75	1·97	·42
16	…	…	…	…	1·98	2·20	·44	1·85	2·08	·44
17	…	…	…	…	2·09	2·33	·46	1·95	2·19	·46
18	…	…	…	…	2·20	2·45	·48	2·05	2·30	·48
19	…	…	…	…	2·31	2·58	·50	2·15	2·41	·50
20	…	…	…	…	2·42	2·70	·52	2·25	2·52	·52

Note.—**Thicknesses given in above Table are *total* thicknesses, including white-metal, if any.**

When double brasses are used the proportions differ slightly from the above, and are approximately,—

Rule 89e, 89f, and 89g. $\frac{D}{G}=1{\cdot}25$; $\frac{K}{H}=1{\cdot}5$; and $\frac{E}{H}={\cdot}47$.

But of course, in this latter case, much depends on the type of crosshead employed, and the side of the jaw should be considered as a cantilever, and the section at S made such that the moment of P does not impose a load of more than 9000 lbs. per square inch, when P is taken as five-eighths of the total effective load on the piston.

SHAFTING.

The resistance of a plain cylindrical shaft to simple twisting is given by,—

Rule 90. $$T={\cdot}196\ D^3\times f,$$

where T is the twisting moment in inch-lbs. ; D the diameter of the shaft in inches ; and f the greatest shearing stress on the material,—in lbs. per square inch.

When the shaft is hollow, the formula becomes,—

Rule 90a. $$T={\cdot}196\ \frac{D^4-d^4}{D}\times f,$$

where D and d are respectively the external and internal diameters.

The values of f should not exceed the following :—

	f.	
	Shafts below 10″ dia.	Shafts above 10″ dia.
Wrought-iron forging, .	9,000	8,000
Mild steel forging, . .	12,000	11,000

For steel of greater strength allow 0·38 × elastic limit in lbs. as the working maximum stress.

The torsional stiffness of shafts may be estimated by the following formulæ (Rankine) :—

Rule 91. For solid shafts $\theta_r=\frac{10{\cdot}2\ Tl}{CD^4}$ nearly.

Rule 91a. For hollow shafts $\theta r=\frac{10{\cdot}2\ Tl}{C(D^4-d^4)}$ nearly.

θ_r being the measure of angular displacement in *radians*.

The actual angular movement in degrees taken as θ°, then

$$\text{For solid shafts } \theta^\circ=\frac{584\ Tl}{CD^4},$$

For hollow shafts $\theta^{\circ} = \frac{584\ Tl}{C(D^4 - d^4)}$,

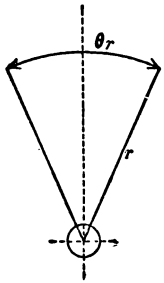


FIG. 14.

θr is the length of arc moved through at radius r; l length of shaft in inches; C constant, values of which are given below; and other symbols, as defined above.

For cast-iron, . .	C = about 2,850,000.
,, wrought-iron, .	C = 8,500,000, to 10,000,000.
,, steel, . . .	C = 10,000,000 to 12,000,000.

Where a shaft is revolving *uniformly*, and transmitting power, the relation between the twisting moment and the horse-power applied is given by,—

Rule 92. $$T = \frac{\text{I.H.P.}}{R} \times 63{,}000.$$

The great number of shafts, however, are turned by steam engines acting through cranks, and do not revolve uniformly,—because the tangential pressure on the crank-pin is constantly varying,—and must therefore be designed to resist the *maximum* twisting moment, instead of the mean given by the above equation.

The rule therefore takes the form,—

Rule 93. $$D^3, \text{ or } \frac{D^4 - d^4}{D}, = \frac{\text{I.H.P.}}{R} \times F.$$

Where F is a co-efficient, the values of which depend on the number and relative positions of cranks, distribution of steam in the cylinders, &c., and are given in Tables L. and La.

In Table L. the cranks marked † are at angles of 120°; those marked ‡ at angles of 90°.

Table L.—Shafts for Screw Engines.

Description of Screw Engines.	Values of F for		
	Thrust, etc., Crank-shafts.	Inter-mediate shaft.	Tail-end shaft.
Single crank cut off in cylinders not less than 0·5,	150	130	175
Two cranks at 90°, cut off in cylinders not less than 0·5	96	84	115
Three cranks at 120°, cut off in cylinders not less than 0·5	83	72	96
Four cranks, balanced triples and quadruples	84	74	96
,, 90° quadruples	86	76	98
,, naval and other high-speed balanced	74	63	85
Turbines, direct drive	70	60	80
,, geared generally	70	60	78
,, naval, in high-speed light craft	37	32	43

Table La.—Shafts for Paddle Engines.

Description of Paddle Engines.	Values of F for		
	Inter-mediate shaft.	Paddle shaft inner journal.	Paddle shaft outer journal.
Single crank cut off in cylinder not less than 0·5,	...	80	100
Two cranks connected by link and equivalent to 90°	...	58	65
Two cranks at 90° with an intermediate shaft	58	50	65
Three cranks at 120°, solid or built up	...	45	60

For ships working in rivers, estuaries, lochs, etc., habitually in smooth water, the value of F may be taken as 80 per cent. of the above.

Hollow Shafting.—Shafting for naval ships is now made throughout hollow in order to get the maximum strength and stiffness with minimum weight.

The internal diameter is usually half the external, so that the saving in weight is 25 per cent., whilst the reduction of strength is only about 6¼ per cent.

"Tail" shafts are usually "set in," at the after end, after boring, so as to reduce the diameter of the hole within and near the propeller boss.

CRANKSHAFTS IN GENERAL.

Crankshafts must be strong enough to bear, together with the maximum torque, the bending and shearing stresses, the magnitudes of which depend on the positions of the bearings. The conditions of loading and support are rather difficult to determine exactly,—depending as they do on the rigidity of the surrounding parts,—but it suffices for most purposes to assume that, in a simple case of overhang, they are represented by the formula,—

Rule 94. $$W = \frac{fz}{l},$$

where W is maximum load on pin, in lbs. ; f the greatest permissible stress on the material in lbs. per square inch (for which *see* page 113) ; l length in inches from mid-length of crank-pin to mid-length of bearing ; and z, for a circular section, is $\cdot 0982\ D^3$.

Where the crank has two arms, and is supported by a bearing on each side, the formula may be taken as,—

Rule 94a. $$W = \frac{6fz}{l},$$

l being the distance, in inches, between the *centres* of the two bearings.

Equivalent Twisting Moment.—When a shaft is subject to simultaneous twisting and bending, the combined stress on any section of it may be measured by calculating what is called the

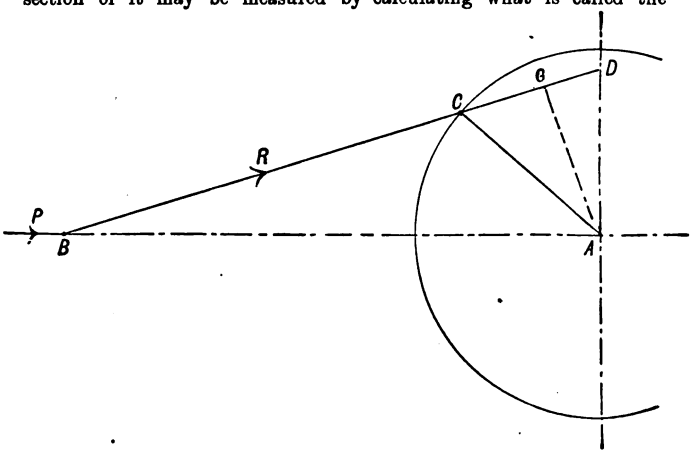


FIG. 15.

equivalent twisting moment; that is, the two stresses may be so combined as to be treated as one twisting stress only, and the size of the shaft calculated accordingly. The formula for combining the stresses is,—

Rule 95. $$T_1 = M + \sqrt{M^2 + T^2}$$

where T is the twisting moment; M the bending moment at any section; and T_1 the equivalent twisting moment.

The shearing stresses on crankshafts, exclusive of those due to torque, are relatively small, and usually allowed for in the value assumed for *f* or F.

Curve of Twisting Moments.—The twisting moment, at any position of the crank, may be determined graphically as on Fig. 15.

Let AB (Fig. 15) be the centre line of the engine, through the cylinder and shaft centres, AC the position of the crank, BC the connecting-rod, and AD a line at right angles to AB. Produce BC (if necessary) to cut the line AD. Then, if P be the effective pressure on the piston, when the crank is in position AC, the twisting moment is P × AD.

Let the twisting moment be determined at intervals of say 10° of angular movement of the crank, so that there will be 18 values obtained for the half revolutions. Draw a line AB (Fig. 16), and

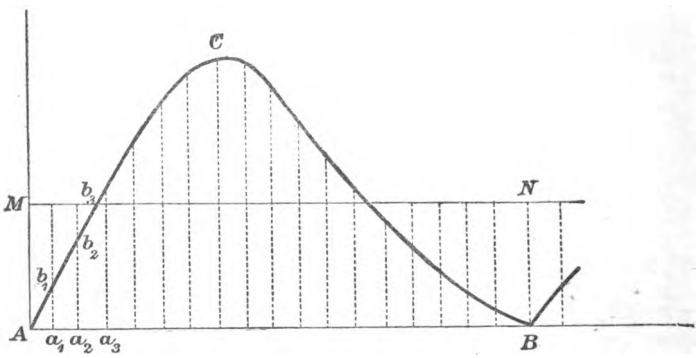


FIG. 16.

divide it into 18 equal parts, Aa_1, a_1a_2, &c.; erect perpendiculars at these points, and cut off parts a_1b_1, a_2b_2, &c., representing to scale the 18 values obtained; and through the points b_1, b_2, &c., draw a curve, which will be the curve of twisting stress on the shaft during one stroke of the piston. By prolonging AB, and going through a similar operation for the second half of the revolution, the curve of stress during the return stroke may be obtained.

If the area enclosed by the curve and the line AB be divided by the length of AB, the quotient (AM in Fig. 16) is the *mean* twisting moment. The value of AM may be calculated by taking a mean of the values of a_1b_1, a_2b_2, &c. Where there are two or more pistons acting on the same shaft, the curve of combined twisting moments is obtained by laying off the ordinates of the second piston, or crank, above or beyond the first curve, which is used as the base line, and so on, care being taken to step, or displace, the various curves in the direction AB, by an amount due to the angles between the cranks.

The polar form of this diagram, which is preferred by some engineers, is obtained by laying off the ordinates of the various curves radially from a circle which represents the crank-path, instead of from a straight line representing the half crank-path unrolled.

The effective pressures on the pistons at each point are obtained from the indicator diagrams in the manner explained on page 13.

The bending moment on a section of the shaft will vary exactly with the pressure on the crank-pin, and to find the maximum equivalent twisting moment on a section, it is only necessary to construct a secondary curve from the formula $T_1 = M + \sqrt{M^2 + T^2}$, between the point of maximum twisting and that at which the pressure on the piston is greatest.

When steam is not cut off in the cylinder before ·4 of the stroke, the maximum load on the piston may be used to calculate the bending moment which is to be combined with the maximum twisting moment.

Effect of Inertia of Reciprocating Parts.—The attention of practical engineers was first drawn to this very interesting subject by Mr Arthur Rigg, in his "Treatise on the Steam Engine," and it has since been treated by various other writers. So far as ordinary marine machinery is concerned, the general effect of this inertia is to equalise the action of the crank in the same manner as a fly-wheel,—the energy expended in accelerating the reciprocating parts during the early part of the stroke being given out again during the latter part, when the steam pressure is lower. When a curve of twisting moments is constructed as described above, and corrected for the effects of inertia, it will generally show that by far the greater part of the steam pressure at the commencement of the stroke is absorbed in producing acceleration, and that the remaining *effective* pressure is but a very small fraction of the whole.

It is not necessary, however, to take these effects into consideration when proportioning the various parts of an ordinary engine, since calculations based on the statical stresses will quite cover all that is necessary to provide for dynamical stresses, except in very extreme cases. For engines of high revolution inertia forces and stresses are not negligible.

Surfaces of Crank-pins and Main Bearings.—If the effective bearing surfaces of pins and journals be considered as equal to

diameter multiplied by length, then the pressures per square inch should not exceed those given in the following Table:—

Table LI.—Maximum Working Pressures on Crank-pins, Main Bearings, &c.

Crank-pins may carry	$\frac{18,000}{\sqrt{Rd}}$	lbs. per sq. in.
Gudgeons ,, ,,	$\frac{22,600}{\sqrt{Rd}}$	,, ,,
Main bearings of cranks	$\frac{12,750}{\sqrt{Rd}}$	,, ,,
Intermediate shaft bearings	$\frac{10,500}{\sqrt{Rd}}$	,, ,,

R being the revolutions per minute. d the diameters in inches.

N.B. With crank pins and main bearings when fitted with Michell pads, the above allowances may be doubled.

The pressures referred to in the above Table are those resulting from the maximum effective loads on the pistons.

Crankshafts of Screw Engines.

Where a shaft has two, three, or more cranks in it, the after crank has not only to resist the stresses imposed by its own cylinder, but also to transmit the torques produced by all cylinders forward of it, so that,—were it not for practical reasons,—the sizes of crank-arms, journals, and pins, might be progressively reduced from after to forward end of shaft.

On the forward journal and forward arm of forward crank, there is little more torque than is necessary to drive the forward eccentrics (if any), and the bending stresses due to half the load on the pin; whilst the after arm of the same crank, and the journal adjoining it, are subject to similar bending stress, and to nearly the whole of the torque due to the load.

If the bending moment in either journal be expressed by $\frac{M}{2}$, then, the equivalent twisting moment in after journal will be,—according to formula given by Rule 95.

$$T_1 = \frac{M}{2} \sqrt{\left(\frac{M}{2}\right)^2 + T^2}.$$

Coming now to the aftermost crank,—if T_2 be the maximum total torque produced by all cylinders forward of it; T_3 the maximum

torque from its own cylinder; and M_3 the corresponding bending moment; then, for the forward journal, the twisting moment is T_2, and the bending moment $\frac{M_3}{2}$, so that the equivalent twisting moment is,—

$$T_4 = \frac{M_3}{2} + \sqrt{\left(\frac{M_3}{2}\right)^2 + T_2^2},$$

whilst, on the after journal, the twisting moment is $T_2 + T_3$, and the bending moment $\frac{M_3}{2}$, so that the equivalent twisting moment is,—

$$T_5 = \frac{M_3}{2} + \sqrt{\left(\frac{M_3}{2}\right)^2 + (T_2 + T_3)^2}.$$

The maximum bending moment, acting in a plane at right angles to the axis of the shaft, at any section of the aftermost crank-arm, distant x from the crank-pin centre, is,—

$$M_5 = \frac{T_2 + T_3}{L} \times x,$$

and for the forward arm of the same crank,—

$$M_4 = \frac{T_2}{L} \times x,$$

where L is the length of the crank in inches.

The bending moment at the mid-length of the pin of a solid or rigidly built-up shaft is,—

Rule 96. $$M_6 = \frac{R \times L}{8},$$

where R is the load on pin, and L the distance between the centres of the main bearings on either side,—the shaft being considered as a continuous girder supported at the centres of the bearings.

In marine crankshafts, where the crank-arms are not less than ·7 of the diameter in thickness, and where the bearings are close to the crank-arms, there is really little or no bending action at the journals.

At and near the ends of the stroke the crank-arms are subjected to a bending moment acting in the plane of the axis of the shaft and has a magnitude,—

$$M_7 = \frac{R \times L_1}{8},$$

where L_1 is the distance from the mid-length of pin to the fore-and-

aft centre of the arm (Fig. 17). Hence the thickness (a) of the forward arm of forward crank must never be less than,—

$$a = \sqrt{\frac{3R \times L_1}{4bf}}.$$

In ordinary practice, however, to secure interchangeability, the crankshafts of merchant steamers are made with all journals of the same diameter, and all crank-arms of the same dimensions; but in

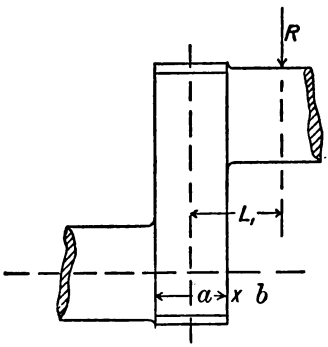


FIG. 17.

light machinery for Naval purposes the forward crank-arms are sometimes reduced in fore-and-aft thickness.

Crank-arms, when forged solid with the shaft, will be found to satisfactorily resist all stresses to which they are subject, if made of the following proportions:—

Rule 97. **Breadth of arm** = 1·1 × Diameter of shaft.

Rule 98. **Thickness of arm** = ·75 × Diameter of shaft.

When for any reason it is not possible to make the arms so thick as ·75 × diameter of shaft, they may be made as thin as ·7, or even, in special cases, ·65; but when this is done, care must be taken to make the breadth such that,—

$$\frac{a \times b^2}{6} = \cdot 196\ D^3 \times \cdot 77 = \cdot 151\ D^3.$$

The most destructive of all stresses to crankshafts are, however, those resulting from the main bearings getting out of line, and the levels of these should therefore be verified every few months, and the shaft lifted and re-bedded whenever any unequal wear is apparent.

The crank-pins of screw engines are nearly always of the same diameter as the shaft journals, and now never less in length than the diameter. The length of the pins, however, will depend on the load per square inch on ($d \times l$) as allowed by scale in Table LI.

The length of shaft journals cannot, as a rule, be less than the diameter, from practical considerations, and should be such as to comply with scale in Table LI.

To ensure equal wear in the main bearings, they should, theoretically, increase in length from forward aft, in proportion to the twisting moment at each journal,—as the turning force at the crank-pin tends to cause the shaft to move round eccentrically within the bearing and so cause the wear ; the after bearing *is* found, in practice, to wear more rapidly than the others, and should not be cut away at the sides, as it has to take the reaction due to the torque of the other cranks as well as that of its own.

Shaft Couplings and Coupling Bolts.—(Rule 99.) When couplings are forged solid with the shaft they should have a thickness of ·28 × diameter of shaft.

Coupling bolts should have little beyond a shearing stress to resist, and their size is therefore given by,—

Rule 100. Area of one bolt $= \dfrac{\cdot 196 D^3}{N \times r}$ or $\dfrac{\cdot 196 \left(\dfrac{D^4 - d^4}{D} \right)}{N \times r}$ *

if the material of the bolts and that of the shaft offer the same resistance to shearing ; D being diameter of shaft, N number of coupling bolts, and r radius at which they are placed.

By way of making some allowance for screwing-up stresses on small bolts, coupling bolts below 1¼-inch diameter may be made 20 per cent. larger than given by above rule.

Where the bolts in any coupling may require to be removed at times, they should be made with a taper of about ¼-inch per foot on the diameters, and be made of harder steel than the shaft.

It is by no means necessary, in large coupling bolts, to make the screwed part equal in diameter to the body of the bolt ; bolts between 2½ and 3½ inches diameter may be reduced ¼-inch, and those above 3½ inches, ½-inch in diameter. The nuts may have a thickness of ·75 to ·8 × diameter of screw.

In two, three, and four throw crankshafts, above 10 inch of diameter, each crank, with its adjacent journals, should be made as a separate forging, and the number and position of coupling bolts should be such that the shafts can be interchanged or reversed as may be desired.

---

* *Note.*—Or to obtain diameter at once,—

$$d = \sqrt{\frac{D^3}{4 N r}} + \cdot 125, \text{ \&c.}$

When cranks are coupled together in this way, a spigot on each, fitting into a recess in its neighbour, materially reduces the strain on the bolts, assists adjustment, and ensures the "truth" of the shafts when bolted together; such spigots may have a diameter of ·5 to ·7 × diameter of shaft, and a length of say ⅜-inch for the projection, and $\frac{7}{16}$-inch for the recess.

Built Crankshafts.—With a view of obtaining sounder and more reliable work, crankshafts are usually built up of separate pieces. Such shafts are generally cheaper than solid ones, but somewhat heavier.

The thickness of the arms or *webs* are usually somewhat less than those of solid shafts. They should, however, not be less than 0·625 the diameter of the adjacent journal. If t is thickness of the metal around the shank hole, and h the thickness of the web, and d the shank of journal, $t^2 \times h$ should not be less than $0{\cdot}12 \times d^3$. These webs are shrunk on to the shanks and crank-pins; there should be a key or keys on the shaft which are generally cylindrical and sometimes screwed into their holes to prevent end play in case the web gets loose.

The diameter of such key should be not less than $0{\cdot}23 \times d + 0{\cdot}2$ inch.

Crankshafts of Paddle Engines.

These may usually be classed under one of the following three heads:—

(*a*) Intermediate shaft type,—in which the outer crank-arms are keyed upon the paddle-shafts, where the inner ones are similarly fixed to an intermediate shaft, the crank-pins being fixed in the inner crank-arms, and, to a certain extent, free in the outer ones (*v.* fig. 18).

(*b*) The type in which the intermediate shaft and its crank-arms are absent, and the crank-pins are fixed in the paddle-shaft crank-arms, and left long enough to allow of the attachment of a link, or sling, which holds the crank-arms at the required angle with one another.

(*c*) The solid shaft,—in which the cranks are forged in one with the shaft—as in the case of a screw engine,—and the paddle-shafts connected with it by solid flange couplings, a little elasticity being obtained sometimes by the interposition of a thickness of leather, by helical springs on the coupling bolts, or by other similar devices. Paddle-engine shafts are now generally built up like those of screw engines with the pins of the same diameter as the journals.

The effect of the first arrangement is to communicate very equable stresses to the paddle-shafts, as the pressures from the crank-pins are always at right angles to the cranks on the paddle-shafts; and, in smooth water, the power of each cylinder is very nearly equally divided between the two wheels, and the bending action on each paddle-shaft never exceeds half that due to its own cylinder, for, when near the dead-points, the bending moment is at its maximum, and is wholly taken by the inner crank-arm, to which the pin is secured. For these

reasons the intermediate shaft must be stronger than the paddle-shafts, when the ship is intended to work in rough water, as it may have to transmit the whole twisting force from one cylinder, and always takes, during certain parts of the revolution, the whole of the bending forces.

Hence, if T be the maximum twisting moment from one piston of a double cylinder paddle engine, and M the maximum bending moment from the same piston, then,—as in the general case (page 126),—the maximum equivalent twisting moment T_1 will be,—

$$T_1 \text{ (on intermediate shaft)} = M + \sqrt{M^2 + T^2}$$

$$\text{and } T_1 \text{ (on paddle-shafts)} = \frac{M}{2} + \sqrt{\left(\frac{M}{2}\right)^2 + T^2}.$$

With the second type of shaft (*b*), the axes of the cylinders are necessarily at an angle with one another, but the cranks are usually so placed that the arrangement is equivalent to one in which the axes are in the same plane and the cranks at right angles.

Each shaft takes the whole of the bending moment from its own cylinder; and,—while usually transmitting half the combined twisting moment—may, in rough water, have to transmit the whole of the twisting moment from one cylinder. The inner journals of these shafts are therefore subject to precisely the same stresses as the intermediate shaft journals in type (*a*), and should be made of the same size, as indicated in Table L.a, page 124.

The maximum pull in the coupling link may be taken as being equal to that in one connecting-rod, but its ordinary value will only be one half of this.

In the third case (that of the solid crankshaft), the cranks will generally be at right angles, and the twisting stresses will be similar to those in the previous cases; the central part transmitting ordinarily half the stress from one piston, but occasionally the whole. The bending stresses in each crank-arm and journal will be those due to half the load on one piston, and may be determined by means of the formulæ previously given.

Paddle-shafts.—The outer end of each paddle-shaft is subjected to a bending moment, which is the resultant of those due to the weight of the wheel, and to the reaction of the water on the floats, and the moment of which (M_1) may be taken as,—

Rule 101. $M_1 = \sqrt{(\text{M of weight})^2 + (\text{M of reaction})^2}$.

The torque at the outer journal of a paddle-shaft is the same as at the inner journal. The outer bearing may have a length of from 1·5 to 2 × diameter of journal, according to service for which the boat is intended, and to weight of wheels.

Overhung cranks.—Fig. 18 shows a pair of overhung cranks as formerly fitted in paddle-wheel engines. The crank-pin is subject

to bending and shearing stresses due to the thrust on the connecting-rod. The maximum bending stress in the pin is close to the face of the crank and is,—

$$M = R \times l,$$

where R is the thrust on the connecting-rod, and l the length of pin from centre of connecting-rod to face of crank.

The diameter of the pin is given by,—

Rule 102. Diameter of Crank-pin $= \sqrt[3]{\frac{R \times l}{f}} \times 10.2,$

where f is the greatest permissible stress on the material, in lbs. per square inch. This rule gives the diameter requisite for strength, but it may be necessary to make the pin larger in order to get sufficient surface to comply with Table LI., page 128.

The crank-arm is to be treated as a lever, so that,—if a is thickness measured parallel to axis of shaft, and b breadth, at a section x inches from the crank-pin,—the bending moment at that section is,—

$$M = R \times x$$

and,
$$\frac{a \times b^2}{6} = \frac{M}{f}$$

$$\text{or } a = \frac{6M}{b^2 \times f}.$$

The bending moment decreases with the distance from the crank-pin, while the shearing force is the same throughout the crank-arm; consequently the section near the crank-pin should have an extra square inch for each 10,000 lbs. of thrust on the connecting-rod, beyond the area necessary to resist the bending moment.

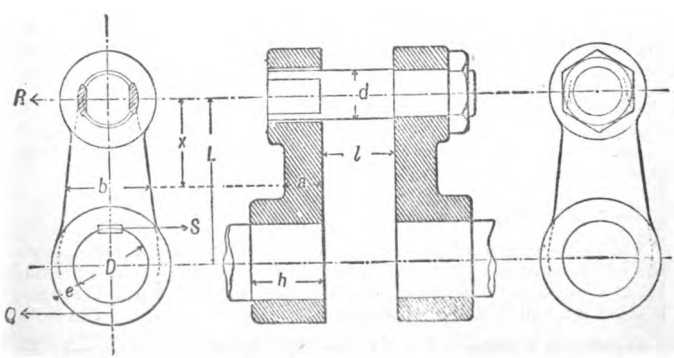


FIG. 18.

The dimensions of the boss to receive the shaft may be determined from,—

Rule 103. $$T = 2R\frac{L}{D+e},$$

where T is the total stress on the section $h \times e$, and the other letters have the meanings shown in Fig. 18. The length h is usually ·75 to 1 × diameter of shaft (**Rule 103a**), and,—when crank and shaft are of the same material,—h and e may have the following relative values:—

When h = D then e = ·35D.
,, h = ·9D ,, e = ·38D.
,, h = ·8D ,, e = ·40D.
,, h = ·7D ,, e = ·41D.

The crank-eye, or boss into which the crank-pin is fitted, should bear the same relation to the pin that the shaft boss does to the shaft.

The diameter of the shaft end, on which the crank is fitted, should be 1·1 × diameter of journal.

Keys for cranks, &c.—These should be made of steel several grades harder than the shaft and crank, and should be of the following proportions:—

Rule 104. **Breadth of Key** $= \frac{D}{4} + ·125.$

Rule 104a. **Thickness of Key** = ·5 × Breadth.

Depth of key-way in shaft, at edge, should be ·17 × Breadth. Some engineers use two (smaller) keys placed 90° apart, in order to obtain at least three bearing points on shaft, and so avoid all risk of "rocking"; and large shafts are sometimes fitted with three keys. Where a lever is placed at some distance from the end of a shaft, so that the key cannot be fitted and driven from the end, it is a good plan to fit a small driving key alongside of the sunk key, as shown in Fig. 19.

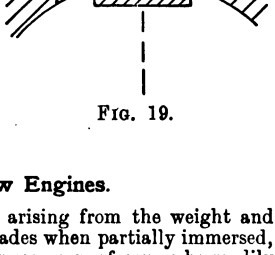


FIG. 19.

Tail Shafts of Screw Engines.

These must resist the bending action arising from the weight and inertia of the propeller, and from the blades when partially immersed, in addition to the torque. The static stresses can of course be readily determined by the rules already given, but the calculation of the stresses resulting from "racing" of the propeller and pitching of ship is too complex a subject to enter on here. Experience has shown, however, that the tail shaft should have a diameter greater than that of the crankshaft; and it is in practice the rule now to make them considerably larger than the crankshaft, since breakage of one may

involve serious consequences. Much also depends on the manner in which the tail-shaft is supported.

Rule 105.—The taper of the end that fits into the propeller boss should not be less than ¾-inch per ft., and is generally 1 inch per ft.

The thread of the large nut that holds the propeller on the shaft is commonly made 4 threads per inch, regardless of the diameter; it should be left-handed when the propeller is right-handed; and *vice versâ*. The nut should be securely locked, preferably by a plate fixed to the after-end of the boss by tap screws.

The propeller should be secured by a feather key of such a size that the pressure per square inch on its bearing sides does not exceed 11,000 per square inch. It generally extends the full length of the boss, but

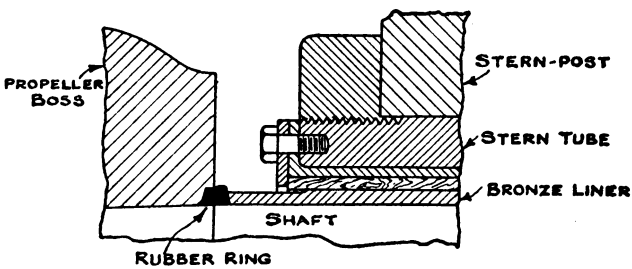


FIG. 20.—Shows an arrangement of after-end of the stern tube, shaft, and screw boss as generally adopted.

sometimes it is only about half the length, being limited to the bearing part before the recess. When the pressure exceeds 11,000 lbs. per square inch with one there should be two keys, as is common practice with Naval ships of high power.

The breadth of the key should be 2 × thickness.

The thickness within the boss when it is of bronze should be 12½ per cent. more than given when the boss is of cast steel or iron.

The propeller boss should have a recess at the fore-end, into which an indiarubber ring of somewhat smaller external diameter is fitted, and against which the shaft liner end is butted and so made watertight.

Rule 106.—Breadth of key = ·22 × largest diameter of shaft + ·25.

Rule 106a.—Thickness of key = ·55 × breadth.

The diameter of the screwed end of the shaft should be sufficiently reduced to allow of the key being fitted in, from the after-end, clear of the thread.

The diameters over the brass casings at the bearings should differ by ⅛ inch, to enable the shaft to be got in and out of the stern-tube more easily. The thickness of brass casing should not be less than,—

Rule 107.—Thickness of brass casings at bearings should be not less than $13\sqrt{D_1}$ in 64ths of an inch; and between the bearings and the propeller boss $10\sqrt{D_1}$, where D_1 is the diameter in inches of the shaft at the after stern bush. The liners should bear on the full length and be shrunk on or pressed on tight.

See Table LV., p. 144.

Board of Trade Rule for Paddle Shafts.

Paddle shafts must not be less in diameter than given by this formula, and in practice are generally larger, especially in oversea ships and tugs.

For compound condensing engines having two or more cylinders and the cranks not "overhung":—

Rule 108. $$S=\sqrt[3]{\frac{C\times P\times D^2}{f(2+r)}} \quad \text{and} \quad P=\frac{f\times S^3}{C\times D}(2+r),$$

where S is the diameter of shaft at journal in inches.
D is the diameter of low-pressure cylinder in inches.
C is the length of crank or half the piston stroke in inches.
P is the pressure load on safety valve + 15 lbs.
f is a factor being 1466 when there are two cranks at an angle of 90°, and 1554 when three cranks at 120°.

Rules for the Shafting of Screw Ships (as established by the British Marine Engineering D. & C. Committee, 1920).

(i.) For the shafts with turbine motors.

Rule 108a. Diameter of intermediate shafts $=\sqrt[3]{\frac{S.H.P.\times F}{R}}$.

S.H.P. is the maximum shaft horse-power (at full speed) when running at R revolutions.

For ocean-going steamers F = 64.
For steamers on service in smooth waters F = 58.

(ii.) For reciprocating steam engines the diameter of the low-pressure cylinder of which is D, and the piston stroke is S, both in inches.

r is the ratio of the L.P. cylinder capacity to that of the H.P.
P is the working boiler pressure.
F_1 is a factor the value of which is as under.

Rule 108b.—Diameter of intermediate shaft $= \sqrt[3]{\frac{D^2 \times S \times P}{F_1(r+2)}}$.

Table LII. Values of F_1 for Various Reciprocating Engines.

No.	General Description of Engines.	Service of Ship.	
		Ocean.	Smooth Water.
a	2 Cranks at 90° Cylinders Compound, Triple, or Quadruple	1,800	2,000
b	2 ,, ,, 180° ,, ,, ,, ,, ,,	1,260	1,400
c	3 ,, ,, 120° ,, ,, ,, ,, ,,	2,000	2,250
d	4 ,, Balanced ,, ,, ,, ,,	2,000	2,250
e	4 ,, at 90° Cylinders Quadruple.	1,900	2,150

The Crank Shaft, the Thrust Shaft (in the way of the thrust collars), and the (inner end of the) *Tube Shaft* shall be not less than,—

Rule 108c.—Diameter = 1·05 × diameter of intermediate shaft.

The shaft carrying the main spur wheel of a gearing arranged drive shall be not less than this at the wheel end between bearings.

Rule 108d.—Diameter = G × diameter of intermediate shaft.
When there are two pinions geared to the spur wheel and the angle between the planes passing through the axes of their spindles is not less than 120°, the value of G is 1·05.
When that angle is less than 120°, or only one, G = 1·1.
Tail Shafts within the stern bush, whether it is in the stern tube or in the brackets of multiple screw ships, shall be not less than,—

Rule 108f.—Diameter of tail shaft end $= d + \frac{P}{K}$,

where d is the diameter of the intermediate shaft as given by Rule 108a or 108b.
P is the diameter of the circle through the screw blade tips.
K is a factor as follows.

N.B.—The above Rules have been approved by the Board of Trade, Lloyd's, and the other Register Societies, and generally taken as substitutes of their older Rules.

Table LIII.—Values of K for Tail Shafts.

Description of Engines.	Service of Ship.	Values of K when	
		The Liner is Continuous.	Liners are Non-Continuous.
Turbines or Electric Motors.	Ocean and Cross Channel.	144	120
Reciprocators generally.	Ocean and Cross Channel (B.T. Nos. 1 and 2).	120	100
Reciprocators, Turbines, and Motors.	Smooth water (B.T. Nos. 3, 4, or 5).	144	120

THRUST BLOCKS.

To find the thrust along the shaft of a screw engine, it is necessary to know the speed of the ship, and the effective horse-power delivered by the screw, that is, S.H.P. minus the power lost by friction, etc., of the screw itself.

The effective horse-power is the power actually employed in propelling the ship, and its relation to the indicated horse-power depends therefore on the combined efficiency of the engines and propeller.

For the purpose of calculating the surface of thrust-collars it is sufficient to assume the E.H.P. ÷ I.H.P. = K when

For turbine engines of best make, ... K = 0·80
,, best fast-running vertical reciprocators with no pumps, . K = 0·77
,, such engines with air pumps, ... K = 0·75
,, good engines of medium speed having air, feed, and bilge pumps, K = 0·71
,, ordinary mercantile engines with such pumps, . . . K = 0·68

Then,—

Rule 109. Indicated Thrust in lbs. $= \frac{\text{I.H.P.} \times 33{,}000}{\text{S} \times 101{\cdot}3} = \frac{\text{I.H.P.} \times 326}{\text{S}}$,

where S is the speed of the ship in knots; and,—

Rule 110. Mean Normal Thrust in lbs. $= \frac{\text{I.H.P.} \times 326 \times \text{K}}{\text{S}}$.

As the thrust varies with the I.H.P., and inversely with the speed (*see* Table LIV.), it may rise considerably above the "mean normal" value, if, from any cause, the speed is reduced without a corresponding decrease in the power, as, for instance, when "thrashing" against a head wind and sea, or when towing.

Rule 111. The pressure per square inch, due to the mean normal thrust, should be about $(2700 \div \sqrt{Rd} + 100)$ lbs., R being revolutions per min. and d diameter of thrust-shaft in inches; for tug boats, it should not exceed 50 lbs.

In practice the pressure is about 50 lbs. in Naval work, and about 60 lbs. in mercantile steamers; with the Michel block (fig. 21) it may be as high as 500 lbs. with a coefficient of friction 0·0015 only. In practice 250 for Naval, 300 lbs. mercantile is general with this.

The relation between depth and thickness of collars on shaft is given by,—

Rule 112. $$\text{Thickness} = \begin{cases} \cdot 25\,(D-d) + \cdot 4 \text{ inch (with small collars),} \\ \cdot 17\,(D-d) + \cdot 4 \text{ inch (with large collars),} \end{cases}$$

where D is diameter over collars, and d diameter between collars,—both in inches.

The spaces between collars on shafts should be as follows:—

Rule 113. When rings in block are solid gun-metal. } $\text{Space} = \cdot 4\,(D-d)$.

Rule 113a. When block is fitted with cast-iron or cast-steel "horse-shoes" faced with brass or white metal, as shown in fig. 22. } Cast-iron,— $\text{Space} = 1 \cdot 8\sqrt{(D-d)} - 1 \cdot 5$. Cast-Steel,— $\text{Space} = 1 \cdot 5\sqrt{(D-d)} - 1 \cdot 5$.

Rule 113b. When rings are cut out of solid metal of block and cap, and entire corrugated surface is covered with uniform layer of white metal. } $\text{Space} = \cdot 4\,(D-d) + \cdot 6$ in.

The number of collars employed is, to some extent, optional. With the Michel thrust block one is sufficient and usual, but for very large power a second can be added to get the necessary surface with a reasonable diameter. With the ordinary horse-shoe thrust it is usual to have one collar for shafts up to 6 inches diameter, and over that an additional collar for each 2 inches of diameter, or such a number as will give the necessary surface with a moderate diameter.

The shaft should be supported and "steadied" in its place so as to bear properly on the horse-shoe collars without any tendency to oscillate and cause unfairness in contact. To attain this there should be a bearing on each side of the block when it is a large and long one, and for small shafts with a short block one will serve the purpose, and it may be in one with the block.

The lubrication must, of course, be certain and plentiful and applied at the right places for efficiency; the collars should really run in an oil-bath so as to be self-lubricative. In the case of the Michel thrust the lubrication is perfect, and consequently with this system the pressure necessary to squeeze out the oil-film interposed between the rubbing surfaces is very high. Large collars of type fig. 22 may be made hollow with advantage and then fitted with water circulation to carry away the heat generated. To facilitate examination and refit the thrust shaft should be a short one.

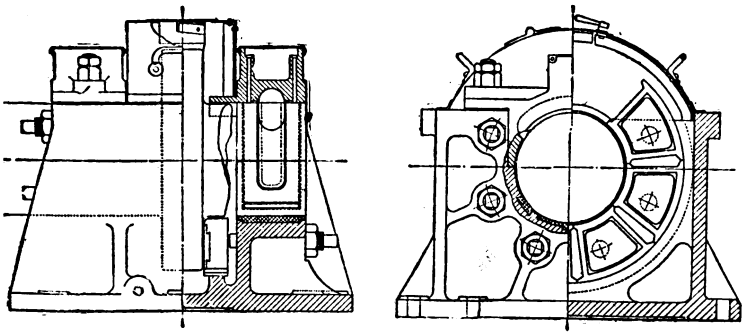
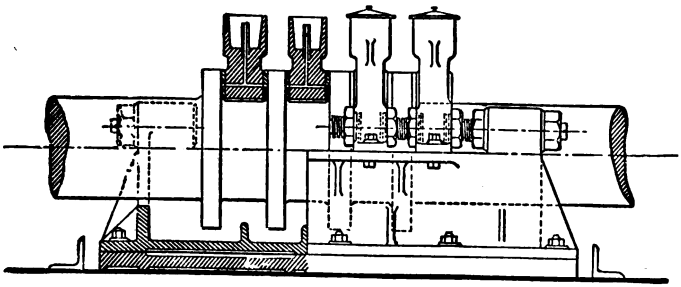


FIG. 21.—Michel Block.



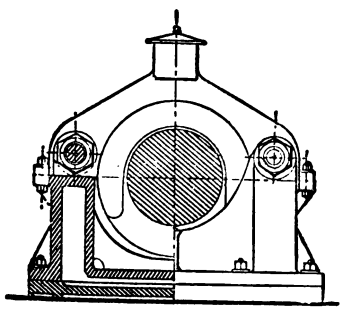


FIG. 22.—Thrust Block.

Thrust-blocks of all types should be carefully scraped and bedded to a perfect bearing on the shaft collars before leaving the workshops.

Rule 114. The number and diameter of holding-down bolts should be such that the shearing stress per square inch, at the bottom of thread due to the mean normal thrust, may be between 1800 and 2200 lbs., when the bolts are unassisted by stops or angle bars riveted to the top plate of the seating, and between 3200 and 3800 lbs. when such stops are fitted. Stops should always be fitted where possible, as, when they are absent, the stress on the bolts is largely a bending stress, owing to the unavoidable presence of a certain thickness of packing.

The sizes of bolts usually employed vary from ⅞ inch to 1½ inch.

Packing should be of cast-iron, *not* of wood.

Table LIV.—Thrust Surfaces per I.H.P. for Various Speeds of Vessels.*

Speed in knots.	Surface necessary to keep pressure 50 lbs. per sq. inch.	Speed in knots.	Surface necessary to keep pressure 50 lbs. per sq. inch.	Speed in knots.	Surface necessary to keep pressure 50 lbs. per sq. inch.	Speed in knots.	Surface necessary to keep pressure 50 lbs. per sq. inch.
	Sq. inch.		Sq. inch.		Sq. inch.		Sq. inch.
8	·540	15	·288	22	·198	29	·150
9	·485	16	·270	23	·190	30	·145
10	·436	17	·255	24	·182	31	·141
11	·396	18	·240	25	·175	32	·137
12	·361	19	·227	26	·168	33	·133
13	·335	20	·216	27	·161	34	·129
14	·312	21	·207	28	·155	35	·125

STERN-TUBES.

In merchant vessels the stern-tube is almost invariably of cast-iron, fitted with a brass bush, or bushes, to carry the lignum-vitæ strips which form the actual bearing surface for the shaft.

In Naval vessels, on the contrary, the stern-tube is almost always of gun-metal, and the bushes are not always fitted, as the lignum-vitæ strips may be fitted into grooves in the tube itself.

In multiple-screw Naval vessels the shaft brackets are fitted with gun-metal bushes, carrying strips of lignum-vitæ,—very similar to those used for the after ends of stern-tubes in merchant vessels; and the forward ends of the stern-tubes are also fitted with long lignum-vitæ bearings, for the purpose of reducing vibration in the unsupported outboard shafts.

The stern-tubes of merchant vessels should be secured in place by a ring nut on the after end, made to screw up against the after face of

* With Michel thrust block allow one-seventh of these surfaces.

the stern-post and draw the tube aft until a collar, formed on it, is close up to forward side of post; the forward end of tube being, of course, jointed to the collision bulkhead in the usual manner.

Naval stern-tubes do not require any fixing at the after end, as they are always surrounded by steel tubes, ½- to ⅝-inch thick, which tie the collision bulkhead to the post, or, in the case of twin-screw ships, to the special stern-tube brackets; they are simply drawn tightly into the post, or brackets, and bolted to the bulkhead at the forward end. When, as occasionally happens, they are put in from the after end, the flange at the stuffing-box end is made separately and screwed on.

Cast-iron stern-tubes should be of the thickness given by,—

Rule 115. $$T = \frac{D}{12} + \cdot 5 \text{ in.},$$

and gun-metal tubes and bushes for shaft-brackets by,—

Rule 115a. $$T = \frac{D}{22} + \cdot 35 \text{ in.},$$

where D is the greatest diameter of shaft, over casings. (*See* Table LV., page 144).

The stern-bush, for either shaft-bracket or stern-tube, should be of such length that the pressure per square inch (taking surface as length × diameter) does not exceed 35 lbs.

Length of bush is now at least 4 × diameter of shaft, over casings.

This rule will be found to limit the pressure to under 30 lbs. per square inch, but it should be in accordance with,—

Rule 116. $$\text{Pressure per sq. inch} = \frac{1250}{\sqrt{Rd}}.$$

The bush at forward end of stern-tube should not be shorter than one diameter for single screw vessels, and, for twin-screw vessels, where there is a length of unsupported shaft outside the stern-tube, the length of this bush should not be less than 1·5 to 1·75 × diameter of shaft.

These bushes in stern-tubes should have a thickness of metal at the back of the lignum-vitæ not less than,—

Rule 116a. $$T_1 = \frac{D}{28} + \cdot 25 \text{ in.},$$

where D is diameter of shaft over casing.

The ribs between the lignum-vitæ strips may have a thickness of ⅜-inch for a shaft 6 to 8 inches in diameter, increasing to ¾-inch for a shaft of 18 inches diameter.

The lignum-vitæ strips themselves may vary from ½-inch × 2 inches to 1 or 1⅛ inch × 4 inches, for shafts of the diameters just mentioned.

The stuffing-box and gland at the inner end of stern-tube may have the following proportions :—

Rule 117.

Depth of stuffing-box $= \frac{D}{2} + 1{\cdot}5$ ins.

Rule 117a.

Amount gland can enter $= \frac{D}{5} + 1$ in.

Rule 117b.

Diameter of stuffing-box $= (1{\cdot}1 \times D) + 1{\cdot}5$ ins. for bushed glands.

Rule 117c.

Diameter of stuffing-box $= (1{\cdot}06 \times D) + 1{\cdot}5$ ins. for solid G.M. glands.

The gland studs may vary from four, 1 inch in diameter, for a shaft of 6 to 8 inches diameter, up to eight, 1¼ inch diameter, for a 16- or 18-inch shaft. If pinion nuts and toothed rings are used, the gland studs must, of course, be made stouter to stand the bending stresses, and should then vary from 1¼ inch to 1⅝ inch.

The following Table shows at a glance the thicknesses of shaft casings, stern-tubes, and stern-bushes, and the dimensions of stern-tube stuffing-boxes, as calculated by Rules 107, 115, 116a, 117 :—

Table LV.—Stern-Tubes and Bushes, etc.

Dia. of shaft under casing. (D_1)	Dia. of shaft over casing. (D) (casing = ·04 + ·2)	Thickness of cast-iron tube. $\frac{D}{12} + {\cdot}5$	Thickness of gun-metal tube or shaft-bracket bush. $\frac{D}{22} + {\cdot}35$	Thickness of tube bushes at back of lignum-vitæ. $\frac{D}{28} + {\cdot}25$	Depth of stuffing box. $\frac{D}{2} + 1{\cdot}5$	Amount gland can enter. $\frac{D}{5} + 1$	Dia. of stuffing box for bushed cast-iron gland. 1·1 D + 1·5	Dia. of Stuffing box for solid gun-metal gland. 1·06 D + 1·5
6	7	1 1/16	11/16	½	5	2⅜	9¼	9
7	8	1 3/16	¾	½	5½	2⅝	10¼	10
8	9⅛	1¼	¾	9/16	6	2⅞	11½	11⅛
9	10¼	1⅜	13/16	⅝	6⅝	3	12¾	12⅜
10	11⅜	1 7/16	⅞	11/16	7⅛	3¼	14	13½
11	12⅜	1 ½	15/16	11/16	7¾	3½	15¼	14¾
12	13½	1⅝	15/16	¾	8¼	3¾	16½	16
13	14½	1¾	1	¾	8⅞	4	17¾	17⅛
14	15⅝	1 13/16	1 1/16	1 1/16	9⅜	4¼	19	18⅜
15	16⅝	1 15/16	1⅛	⅞	10	4⅜	20¼	19½
16	17¾	2	1 3/16	⅞	10½	4⅝	21⅜	20½
17	18¾	2⅛	1 5/16	15/16	11	4⅞	22½	21¾
18	19¾	2 5/16	1¼	1	11⅝	5	23¾	23
19	20⅞	2¼	1 5/16	1	12⅛	5¼	25	24⅛
20	21⅞	2⅜	1⅜	1 1/16	12¾	5½	26¼	25⅜

In designing gun-metal stern-tubes, shaft casings, &c., great care must be taken to avoid, as far as possible, all shoulders or steps in the longitudinal section, as the castings set so rapidly, and consequently grip the core so quickly, that everything tending to hinder the longitudinal contraction is a source of danger; want of attention to this will probably cause a long tube to be torn into two or three separate pieces in cooling. The middle portions of Naval stern-tubes are commonly made about 30 per cent. thinner than the ends or portions containing the bearings. In torpedo boats, &c., the tube-shafts seldom have any casings, and run in white metal bearings.

MAIN BEARINGS.

The surfaces and lengths of main bearings have already been dealt with.

Main Bearing Bolts.—These should be of "mild" steel, with iron nuts, and, to allow for variations in adjustment, &c., should be proportioned as follows:—

Rule 118. When there are two main bearings to each crank, and two bolts to each cap,—assume that each bolt must carry one-third of maximum effective load on piston, and select the proper diameter from Table XLIII., page 103.

Rule 118a.—When there are two bearings on each crank, but four bolts to each cap,—assume that each bolt must carry one-fifth of maximum effective load on piston, and select diameter from same Table.

Rule 118b.—Where there is only one long bearing with four cap bolts, between two cranks of a triple engine (cranks at 120°),—assume that each bolt must carry one-fourth of maximum effective load on piston and select diameter as before.

Rule 118c.—When the crank is overhung, as in various types of paddle-engine, and the whole thrust of the piston-rod, at the ends of the stroke, is resisted by two cap bolts,—call the thrust P; the distance from centre of piston-rod to centre of main bearing bolts pw; and the distance from centre of piston-rod to centre of outer bearing (in case of paddle-shaft), or of bearing at other end (in case of intermediate shaft) pf. Then (fig. 23),—

$$\text{Stress on main bearing bolts} = P \times \frac{pf}{wf}$$

and each bolt must carry one-half of this.

Rule 118d. When the conditions are similar to (c), but there are four bolts in each cap,—assume that each bolt must carry one-third of total load on cap, and select suitable diameter from Table XLIII., page 103.

In cases (c) and (d) the stresses on the main bearing bolts due to the weight and reaction of the wheels should be calculated on the principle indicated above, and allowed for if necessary; their magnitude of course depends very greatly on the length of the paddle-shaft.

Main Bearing Caps or Keeps.—In order to allow for variations

in adjustment, either unnoticed or such as occur when a bearing warms up and has to be slacked, the loads on main bearing caps may be assumed to be as follows:—

Rule 119.

Two bearings to each crank (cases 118 and 118a) . . . . . $\frac{2}{3}$ × **max. effect. load on piston.**

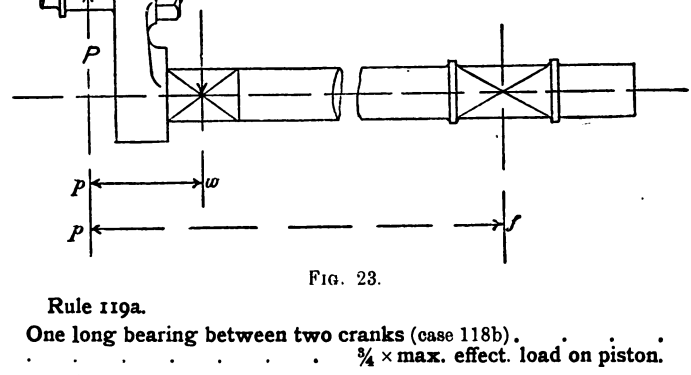


FIG. 23.

Rule 119a.

One long bearing between two cranks (case 118b). . . . . $\frac{3}{4}$ × **max. effect. load on piston.**

Rule 119b.

One bearing to each crank (cases 118c and 118d) . . . . . $\frac{pf}{wf}$ × **max. effect. load on piston.**

The load on a main bearing cap is neither a single central load, nor is it a uniformly distributed load, though probably nearer the latter than the former. If suitable values for the working stresses are employed, the most generally convenient formula is that for a beam supported at ends, and uniformly loaded, viz.:—

Rule 120. $$w = \frac{8fz}{l^2},$$

where w = load in lbs. per inch of span (l);

l = span, or distance from centre of cap bolt to centre of cap bolt;

z = Modulus of section $\left(= \frac{\text{breadth} \times \text{depth}^2}{6} \right.$ for a rectangular section; for other sections see Table CXXXVII.);

and f has the following values:—

Flat-backed brasses and caps	Cast-iron	2,500
	Wrought-iron or "mild" steel	9,000
	Cast-steel	10,000
Half-round brasses and caps	Cast-iron	3,000
	Wrought-iron or "mild" steel	10,000
	Cast-steel	11,000

When the caps are made of cast-steel the section at the centre may, with advantage, be made to approximate to that of a "channel" bar by employing two deep external ribs; but in this case the value of f must not exceed 6000. The values given above for cast-steel are no doubt rather low, but there is,—in the present state of the manufacture,—always some risk of unsoundness, blow-holes, &c.

In estimating the breadths of caps care should be taken to deduct the breadths, or diameters, of all hand holes, oil holes, &c.

Main Bearing Brasses.—For small engines, auxiliary engines, &c., the inner or under brass may be of a semi-octagonal form, and the outer or upper one flat on the back; but in larger engines the under brass at least should be of the "half-round" type, and, when the framing can be arranged to allow of it, it is both cheaper and better to make the upper one "half-round" also. Some engineers, however, prefer the square recess with the flat bottom brass which can be packed up to keep the shaft bearings in true line when required.

The over-all thickness of the brasses (*i.e.* the thickness including white metal) should be as follows:—

Rule 121. Flat-backed brasses $\quad \text{Thickness} = \frac{D}{9} + \cdot 3\text{-inch.}$

Rule 121a. Round brasses . . $\quad \text{Thickness} = \frac{D}{10} + \cdot 25\text{-inch.}$

Rule 121b. White metal . . . $\quad \text{Thickness} = \cdot 02\,D + \cdot 125$ inch, where D is diameter of journal in inches.

The recesses in the brasses should be carefully tinned before the white metal is run in, and the practice of hammering the white metal to consolidate it should be avoided.

In large engines of the mercantile marine the shells for white metal are commonly and are better made of strong cast-iron, and sometimes the caps and shell upper half are cast in one.

When no white metal is used the thickness of brass at crown should be,—

Rule 122.

$$\text{Thickness of brass (when no white metal)} = \begin{cases} \frac{D}{10} + \cdot 15 & \text{when flat-backed.} \\ \frac{D}{11} + \cdot 12 & \text{when round.} \end{cases}$$

Brasses should never overhang the frames or caps by more than "thickness of brass," as given by above rules, at each end.

See Table XLIX. for the proportions given by the above rules.

The tendency of brasses to close, after being warm, and grip the shaft, should be provided against either by securing them to the frames by screws or bolts, or, a simpler and better way, by H section liner strips.

In large main bearings it is very useful to have the lock-rings of the nuts graduated, in order that, after being slacked back, the nuts may be returned exactly to the old positions.

FRAMINGS.

The section of the girder that carries the main bearing is usually either of the "H" or "box" type. For small engines either the H or open bottom box is the best type, and for large engines the box, —if the frame is of cast-iron,—or the H,—if cast-steel is used.

In the case of vertical screw engines for Naval purposes this girder is sometimes cut down to the very slenderest proportions, so far as the engine builder is concerned, and is built of steel plates and angles as a part of the engine seating. In oscillating paddle-engines, on the other hand, the girder receives no support from any seating, and must be strong enough to deal with all stresses communicated to it from the bearing. Between these two extremes almost every possible intermediate case is found in practice, and the designer must meet them by using one or other of the general formulæ for girders, and must use his judgment in deciding what co-efficients best express the conditions of loading, supporting or fixing, &c.,—not forgetting that the material is subject to alternate tensile and compressive stresses.

In the worst case (that of the oscillating paddle-engine) the girder should be taken as *supported* only, and the load considered as a single central one, and the values of f may be 3000 for cast-iron and 8000 for cast-steel. In this particular example there are frequently other stresses,—due to the unsymmetrical positions of the columns or pillars with regard to the bearings,—which must also be allowed for.

In vertical screw engines there are stresses on these girders communicated from the columns, and due to the thrusts on the guides, and also to the weight of the cylinders, &c., when the ship is rolling.

In paddle-wheel vessels there are also (as previously mentioned) stresses due to the weights and reactions of the wheels, &c.

Very valuable additional strength and stiffness may sometimes be obtained by using a forged or cast-steel cap with lips at the ends which prevent the springing open of the gap containing the brasses, or, spigots or projections may be used which will prevent it either opening or closing.

COLUMNS.

The tensile stresses on the material of columns, and also on the bolts connecting them to the cylinders have been dealt with under the heading "Column feet and Bolts," page 97; but in vertical engines

there are also bending stresses due to the thrusts on the guides, and to the weight of the cylinders, pistons, valves, rods, &c., when the ship is rolling. These may all be dealt with by means of the formula for a beam fixed at one end and loaded at the other, viz. :—

$$W = \frac{fz}{l},$$

and need not therefore be considered in detail here.

The bolts attaching the column feet to the bed-plate or frame must of course be strong enough to carry the direct tensile stress, plus the stress caused by the tendency of the column to cant over on one edge of its base; and it must be noticed that only about one-fourth of the bolts in each column foot are, on an average, able to offer resistance to the latter stress.

When weight is of great importance, as in Naval work, and framings are made of the lightest possible sections, special stresses,—such as those resulting from the action of the reversing engine, &c.,—must be carefully considered and allowed for.

CONDENSERS.

1. Jet Condensers.

The Capacity of a jet condenser should not be less than one-fourth that of the cylinder, or cylinders, exhausting into it, and need not be more than one-half,—unless the engine is a very quick-running one: one-third the capacity of cylinder is generally sufficient.

The Form of a jet condenser is not a matter of much consequence, and depends, to some extent, on the type and design of engine to which it is fitted. The inlet for steam should be high enough to prevent water getting into the cylinders, and the lower portion should be so shaped that the water will all drain to the air-pump.

The Position of the Jet Pipe or nozzle, and the form of the delivery openings in it depend so much on the form of the condenser that no definite rule can be given; the nozzle pipe may have a number of small holes drilled in it, or the water may issue from transverse or longitudinal saw-cuts or slots, or may all issue in a sheet, from the end of the pipe, through a nozzle resembling a section of pipe which has been flattened and nearly closed.

The Quantity of Injection Water depends on the weight of steam to be condensed, and on its heat, and,—to ascertain the exact quantity,—the temperature of the injection water, and the required temperature of hot-well must be known.

The vacuum with this type of condenser rarely exceeds 25 inches, and is more commonly 24 inches,—which corresponds to a temperature of about 140°; the temperature of the hot-well varies, in practice, from 110° to 130°.

The number of pounds of injection water (Q) necessary to condense one pound of steam is given by,—

Rule 123. $$Q = \frac{1114 + (\cdot 3 \times T_S) - T_H}{T_H - T_W},$$

where T_S = temperature of steam at exhaust.*
T_H = ,, of hot-well.
T_W = ,, of cooling water.

Rule 124. It is usual to make an allowance of injection water of from 27 to 30 times the weight of steam to be condensed, for vessels in temperate climates, to 30 to 35 times for the tropics.

The relation of weight of steam to volume, to temperature, and to release pressure is given in the following Table:—

Table LVI.—Weight, Pressure, and Temperature of Steam of Low Pressure.

Pressure in		Temperature	Latent Heat	Volume of one Pound of Steam.	Pressure in		Temperature.	Latent Heat	Volume of one Pound of Steam.
Pounds Absolute.	Vacuum Ins.	F°.	F°.	cub. ft.	Pounds Absolute.	Vacuum Ins.	F°.	F°.	cub. ft.
0·3	29·4	67·5	1070	1067	1·7	26·6	120·3	1029	200
0·4	29·2	74·0	1063	800	1·8	26·4	122·4	1028	190
0·5	29·0	80·0	1058	640	1·9	26·2	124·6	1026	181
0·6	28·8	85·5	1054	535	2·0	26·0	126·7	1025	172
0·7	28·6	90·4	1051	461	2·1	25·8	128·6	1024	165
0·8	28·4	94·5	1048	410	2·2	25·6	130·4	1022	158
0·9	28·2	98·5	1045	367	2·3	25·4	132·2	1021	152
1·0	28·0	102·0	1042	333	2·4	25·2	134·0	1020	145
1·1	27·8	105·0	1040	306	2·5	25·0	135·6	1019	140
1·2	27·6	108·0	1038	282	2·6	24·8	136·9	1018	135
1·3	27·4	111·0	1036	260	2·7	24·6	138·2	1017	130
1·4	27·2	113·7	1034	240	2·8	24·4	139·6	1016	125
1·5	27·0	116·0	1033	225	2·9	24·2	141·0	1015	121
1·6	26·8	118·2	1031	212	3·0	24·0	142·2	1014	118

The Area of Injection Orifice, and size of pipes, is governed by the head of water, the vacuum, and the length of piping and number

* Strictly, *absolute* temperatures (Fahrenheit temperatures+461°) should be used in this and all similar calculations.

of bends, &c., or, in other words, by the equivalent head at the condenser.

Neglecting the resistance to flow at the orifice, and in the pipes and passages, the velocity at the condenser may be found as follows :—

Let h be the head, in feet, above the valve on the condenser ; p, the pressure in the condenser in lbs. per sq. inch; h_1 the equivalent head; and g, gravity. Then,—

$$h_1 = h + (15 - p)\ 2{\cdot}3,$$

and velocity in feet per second is $\sqrt{2\ gh_1}$ or $8\sqrt{h_1}$.

Rule 125. In practice, owing to the loss of head resulting from resistances at valves, and in pipes, &c., the actual velocity is only about half that given by the above rule; hence, in designing, it is usual to calculate on a velocity of only 25 feet per second for shallow draught steamers, and 30 feet per second for deeper ones.

The following more concise expressions are derived from the above rules,—

Rule 126.

$$\left.\begin{matrix}\text{Area of orifice} \\ \text{in sq. ins.}\end{matrix}\right\} = \frac{\text{No. of c. ft. of injection water per minute}}{\text{10·4 to 12·5 according to circumstances}}.$$

Rule 126a.

$$\left.\begin{matrix}\text{or Area of ori-} \\ \text{fice in sq. ins.}\end{matrix}\right\} = \frac{\text{Weight of injection water in lbs. per min.}}{\text{650 to 780 according to circumstances}}.$$

A snifting or overflow valve, held on its seat by atmospheric pressure only, should be provided, to prevent pressure or an undue accumulation of water in the condenser: its diameter should be the same as that of the injection valve.

2. Surface Condensers.

These are now supplied to all classes of vessels both for economy in working and to obtain the high vacuum so desirable when the maximum power is required from an engine, and so necessary to the turbine for high efficiency as well as maximum output. The saving in fuel alone as against the common condenser averages 15 per cent., and may be as much as 20.

The benefit of high vacuum is shown by the relative steam consumptions per horse-power, both with turbines and reciprocators, determined by the experiments of Sir C. Parsons and Professor Weighton.

Table LVII.—Effect of Vacuum on Steam Consumption in lbs. per I.H.P. in a Turbine of 1000 H.P., and in a Quadruple and a Triple Expansion Experimental Reciprocator of 200 I.H.P.

Vacuum . . . Ins.	20	22	24	26	27	28	29
A turbine	19·2	18·1	16·9	15·6	14·8	13·9	13·0
A quadruple reciprocator	16·7	16·1	15·5	15·0	14·7	14·5	14·3
A triple ,,	14·8	14·35	14·05	13·90	13·8	13·77	13·76

The surface condenser is itself more costly than a common one, and requires a separate air pump and water pump against the single pump combining the services with a jet condenser. The extra cost, however, is quite trifling compared with the advantages.

The form of a Surface Condenser is not of great consequence, but it is necessary that it shall be such that the economy of cost of construction does not reduce its efficiency, nor should it be that to gain a small increase in efficiency the cost is greatly increased. In a general way the cylindrical body is convenient, cheap to manufacture, and capable of an arrangement of tubes and baffles, whereby a high efficiency is attained in practical working. It is very light, requires no stays to stiffen it, and can be fixed in any convenient place. Mr Weir's condenser, known as the *Unaflow*, is heart-shaped or approximating to a spherical triangle in cross-section, generally placed vertically with the apex at the bottom, so that the steam expands over a large surface on entering the upper part and is caused to pass over the whole of the tubes below so that every square foot is active. The condensed water drops to the sides and drains to the bottom.

Mr Morison's contraflow system effects the same end in a body of different shape; in his condenser there is the same tapering of the body of tubes, and the steam is forced to pass over the whole, the water, however, is collected and led away to the air pump by special means provided. Mr Morison adopts a similar form of barrel, but more like the section of an egg than that of Mr Weir.

Cooling Surface is now entirely of brass tube, and the quantity depends on the requirements of each particular case. If high vacuum is desired the surface must be somewhat more than if a lower amount will do; but the temperature of the cooling water is the most serious factor, especially for high vacuum. Sea-water in several parts of the tropics is 85° F. in summer time, and 80° F. is common. In the North Atlantic generally and about the British coasts 60° F. is the usual summer temperature, and 65° F. a maximum. The temperature of steam or vapour at a pressure of 1 lb. is no more than 102° F., so that with a good supply of cooling water 28 inches of vacuum can be maintained, but 29 inches is impossible in the tropics, because the temperature of vapour is then only 80° F.

Table LVIII.—Summer Temperature of Sea-Water at Various Parts of the World.

Locality.	Temperature.	Locality.	Temperature.
	F°.		F°.
North Sea, off Drontheim, .	60	Indian Ocean, Kurrachi, .	85
,, ,, Bergen, .	63	,, near Colombo,	85
,, ,, Gothenburg, .	61	,, ,, Bombay,	84
Baltic Sea, ,, Stettin, .	64	,, ,, Singapore,	86
,, ,, Stockholm, .	50	,, ,, Penang, .	87
,, ,, Kronstadt, .	66	China Seas, near Hong Kong,	83
Thames, Gravesend, . .	64	,, ,, Woosung, .	69
Scheldt, Antwerp, . .	66	,, ,, Shanghai, .	64
English Channel, . .	62	,, ,, Kobe, .	67
Portuguese Coast, . .	72	Atlantic Ocean, off Lisbon, .	72
Mediterranean, Gibraltar, .	72	,, ,, Madeira,	76
,, Marseilles, .	70	,, ,, Azores, .	76
,, Malta, .	75	,, ,, Cape Verde	81
,, Port Said, .	76	,, North, in places,	73
Black Sea, near Bosphorus, .	74	,, off Ireland, .	66
Red Sea, ,, Suez, . .	87	,, ,, Boston, U.S.A.	65
,, ,, Middle, .	90	Gulf of Mexico, W. Indies, .	83
,, ,, Aden, . .	87	S. America, Buenos Aires, .	71
Indian Ocean, West, . .	85		

The Cooling Surface should in a general way be in proportion to the weight of steam to be condensed. Professor Weighton with a contraflow condenser deposited 33 lbs. per sq. ft. per hour, and under very favourable conditions as much as 40 lbs. has been got. In Destroyers on trial, when new and the surface quite clean, 28 to 30 lbs. can be condensed. Mr Weir has been able to get 35 lbs. with a vacuum of 27·3 ins. Allowing for the surface losing efficiency after service, it may be taken that a square foot can be relied on to condense 20 lbs. with a liberal amount of cooling water. If the steam consumption of a triple engine be taken at 16 lbs. per I.H.P., then : (Rule 126) surface per I.H.P. $= \frac{16}{20} = 0{\cdot}8$ sq. ft. In the same way for a turbine with a consumption of 13 lbs. the allowance should be : (Rule 126a) $\frac{13}{20}$ or 0·65 sq. ft. The following table may be taken as good practice under the conditions named. It may be said, however, that the ordinary cargo steamer with reciprocators, going to all parts of the world, has 1 sq. ft. of cooling surface per I.H.P., and this is sufficient for a good vacuum anywhere.

Table LIX.--Cooling Surface. Allowance per I.H.P.

Description of Engines.	Home Waters.	Tropics.
Triple Compound, Express Steamers, .	0·80 sq. ft.	1·23 sq. ft.
,, ,, Economic Type, .	0·70 ,,	1·06 ,,
Quadruple Compound, Economic Type,	0·65 ,,	0·95 ,,
Turbine driven, Ordinary Type, . .	0·65 ,,	1·10 ,,
,, ,, Express Type, . .	0·80 ,,	1·25 ,,

In Naval ships; where full speed is only attained at intervals for a short time and the condensers of which are always clean, an allowance of half a square foot suffices for Destroyers and other very light high-speed craft, and for the ordinary cruiser and battleship with turbines 0·75 sq. ft. S.H.P. at full power. There is, however, a limitation to the condensing capacity of cooling surface if the supply of water is restricted. The total length of tube through which the water flows should not exceed 400 diameters in the temperate zone for high surface efficiency: 20 lbs. can then be relied on as the rate of condensation, but with so long a flow there will be a serious reduction in the tropics. If, therefore, high vacuum and good rate of condensation is required, with the cooling water 75° to 80° F., the traverse should not be more than is given by the following rule:—

Rule 127. Traverse of cooling water $= 400 \times \frac{6}{Q} \times \frac{d}{12}$,

Q is the quantity of water required to be condensed per square foot of surface per hour, d is the external diameter of tube in inches.

The traverse should not be more than 330 diameters of tube for ships visiting the tropics and requiring high vacuum.

It was usual for the cooling water to travel three times through the tubes; now, with tubes of considerable length, twice through suffices. The flow over the cooling surface should be rapid for good results, and is often at the rate of 300 ft. per minute, this, however, is high for small tubes and requires large power to force it through; 200 ft. per minute is a good rate. The coldest water must be applied to the coolest part of the condenser, so as to maintain the efficiency of the surface as uniform as possible and to get the best results from the whole; hence the cooling water should be admitted at the bottom of the condenser and discharged from the top, where the difference in temperature from the incoming steam is not great but still sufficient to condense a portion. The difference in temperature at discharge for economic working will be about 5° F., and in the tropics even less.

The quantity of cooling water can be estimated by the following formula:—

Rule 128. $Q = (T_1 + L) - T_3 \div (T_2 - T_0) = \frac{1114 + 0{\cdot}3\,T_1 - T_3}{T_2 - T_0}$,

T_1 is the temperature of the steam entering the condenser, and
L its latent heat;
T_0 the temperature of the cooling water at entry, and
Q the quantity of the cooling water at entry;
T_2 the temperature of cooling water on leaving the condenser,
T_3 the temperature of the hot-well.

The following table gives the ratio of least cooling water to the steam condensed at different temperatures of the sea.

Table LX.—Ratio of Cooling Water to Steam Condensed.

Vacuum . Ins.	25·0	25·5	26·0	26·5	27·0	27·5	28·0	28·5	29·0	29·3
Sea water, 50° F.	13·9	14·7	15·4	16·4	18·2	20·1	22·8	28·0	40·4	65·0
,, 60° F.	16·0	17·1	18·1	19·5	22·1	24·8	29·0	38·0	64·3	158
,, 70° F.	18·9	20·5	21·8	24 0	27·9	32·3	40·0	66·3	156	...
,, 80° F.	23·1	25·5	27·6	31·0	37·9	46·3	63·0	133	...	...
,, 85° F.	26·0	29·1	32·8	40·0	50·5	61·5	153	...	...	...

This shows what in practice is the least weight of cooling water required when the pressure at exhaust is 12 lbs. absolute.

Modern air-pumps can maintain a vacuum of 29 inches, but by the above the quantity of water at 70° F. is enormous, and at 80° F. is impossible; the highest vacuum at that temperature being 28·8, and then the ratio is no less than 270. The highest vacuum with water at 70° will be 29·1 inches, with a ratio of 220. In temperate zones with sea-water at 60° F., 29·4 inches can be maintained by passing 280 lbs. for each pound of condensate. In everyday practice, and the condenser not too clean, 10 to 20 per cent. more water may be required than given in the table.

Condenser tubes are made of brass formerly consisting of 70 per cent. of best selected copper and 30 of best Silesian zinc. The Admiralty composition consists of 29 of zinc and 1 of tin. The British Engineering Standards Association specifies the following:—

Standard condenser tubes to be $\frac{5}{8}$ or $\frac{3}{4}$ in external diameter and 18 L.S.G. thick, made of an alloy of copper and zinc, as follows:—

For tubes to contain not less than 70 per cent. of metallic copper, and not more than a total of 0·75 per cent. of materials other than copper and zinc, except if so specified, 1 per cent. of tin and 1½ to 2 of lead.

For screwed glands 60 of copper, 0·75 impurities and 1 of tin and 1½ to 2 of lead if so specified, the remainder being zinc.

Length unsupported.—(Rule 129) When tubes are secured at the tube-plates by screw ferrules and packing, the unsupported length

should not exceed 100 diameters; or, when they are held by tightly fitting ferrules, 120 diameters. If the tubes are longer than this, they should be supported by intermediate diaphragms of rolled brass, from ⅜-inch to ½-inch in thickness, according to size; but care must be taken, in arranging the design, that these diaphragm plates do not interfere with the free flow of the steam to all parts of the condenser.

Tube plates.—These should be of rolled brass of the following thicknesses:—

Rule 130.

When wood ferrules are used . . 1·5 × diameter of tube.

„ screw glands „ . . 1·1 × „ „

"Diameter of tube" signifies, in all cases, *external* diameter.

Fig. 24 shows the method of securing a tube by a wood ferrule, and fig. 25 by means of a screw gland and packing; the dimensions on the figures give the proportions adopted by B.E.S.A. for ¾-inch tubes.

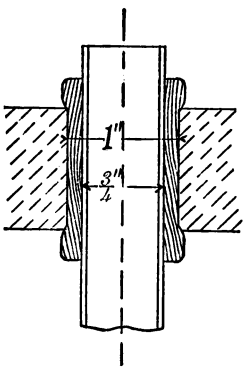


FIG. 24.

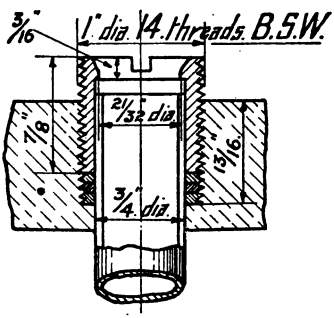


FIG. 25.

Spacing of Tubes.—The tubes should be spaced zigzag, and equilaterally (*i.e.* the centre of any tube should be at the same distance from the centre of each of the immediately surrounding tubes). The pitch may be as follows:—

Rule 131.

When wood ferrules are used—Dia. of hole for ferrule + ¼ in.

When screw glands are used—Extreme dia. of gland + ⅛ to 5/16 in.

The following Table will be of assistance in making rough calculations for size of condenser:—

Table LXI.—Size of Surface Condenser.

External diameter of tube—¾ inch.					
Pitch of tubes, . .	1⅛"	1 5/32"	1 3/16"	1 7/32"	1¼"
Number of tubes per sq. ft. of tube plate, ...	136	128	122	116	110
Cooling surface per cubic ft. of tubes' space, ...	26·7 sq. ft.	25·1 sq. ft.	23·9 sq. ft.	22·7 sq. ft.	21·5 sq. ft.

The surface of 1 foot of ¾-inch tube is ·1963 sq. ft., and of ⅝-inch is ·1637 sq. ft.

See also Table of Surfaces of Tubes, Table CLXI.

Application of Cooling Water.—The simplest method of applying the cooling water is to allow it to run direct from the sea into the condenser, and then to pump it overboard from the condenser; where this cannot be conveniently arranged, and the water is forced through the condenser by a reciprocating pump, a large air-vessel should be provided as near to the pump as possible, to guard against shock from "racing" or sudden starting.

The centrifugal pumps, invariably employed for circulating purposes on ships of war and in passenger steamers and in all ships of large power, suck from the sea and drive the water through the condensers, but they are of course incapable of communicating any shock to the structure.

In order to reduce the shock from reciprocating circulating pumps, some engineers fit a screw-down by-pass valve which admits air when opened more or less, whenever "racing" occurs.

In merchant steamers the condenser was commonly a part of the framing, and made of cast iron; its strength of course depended largely on its form and position, and had little relation to the stresses resulting from its action as condenser.

In war vessels, on the other hand, the condenser never forms any part of the framing; it was commonly cylindrical in form, made of naval brass or gun-metal, and strong enough to bear an internal test pressure of 30 lbs. per sq. inch. It is now in ships generally of sheet steel, heart-shaped in transverse section and having cast-iron water-ways and covers at the ends.

Condensers for Destroyers, &c., up to about 5 feet diameter, usually had the barrels made of sheet brass (8 to 6 L.S.G.), with flanges and stiffening rings riveted and soldered on.

Manholes, sight-holes, mud-holes, and air-cocks should always be provided; in Naval condensers fittings for introducing soda, and for boiling out, are always fitted in addition.

AIR-PUMPS.

Wherever practicable these should be vertical, and single-acting, since this type is in every way the most efficient and satisfactory. Centrifugal pumps are now specially designed and act well as air-pumps in conjunction with ejectors for producing vacuum. *The Kinetic Air-Pump System* is now taking the place of reciprocating pumps for independent condensers for auxiliary machinery.

In vertical screw engines the air-pump should be driven by levers and links from the L.P. crosshead or gudgeon; when driven direct from the piston or crosshead the speed is objectionably high, and the pump is not usually convenient of access. On the other hand, the clearance space is relatively small and it has no gear or parts requiring to be lubricated.

In twin-screw engine installations when the screws turn outwards the air-pump is in front of the engine, worked by levers in the usual way, the condenser being near the ship's side. When the screws turn inwards the pump is at the back, as with single-screw engines. With turbines the condensers are often beside the L.P. member, and the air-pumps being driven by independent motors are placed in any suitable place contiguous.

In paddle-engines the air-pumps are placed in different positions, and driven in different ways according to the type of engine. In oscillating engines they are usually inclined, and driven by connecting-rods from a crank formed in the middle of the intermediate shaft, or by an eccentric fixed in the same position; but are sometimes placed vertically, and driven through a lever or beam. In almost all other types of paddle engine they can be placed vertically without difficulty, and may be driven as found most convenient.

Size of Air-Pump.—Whether for jet or surface condenser, the quantity of water to be removed by the air-pump can be easily determined by the rules given under "Condenser," but the quantity of air is variable and cannot be calculated; and it is therefore necessary, especially in the case of surface condensing engines, to rely more on experience than on theoretical considerations.

The Capacity of Air-Pump depends on the amount of water passing through it from the condenser, inasmuch as the air to be abstracted will vary with the amount of feed water. If a condenser is absolutely tight, the engine glands leak no air, and the circuit of the feed water is closed so that no fresh air comes in contact with it, a very small air-pump suffices to maintain a good vacuum when once formed. But these are conditions not to be counted on in practice; moreover, it is necessary, especially in some services, to have a good vacuum very quickly after the pump starts. The height of the vacuum, or the rarefaction of the condenser, does not depend on the size of the air-

pump, but on its fitness; for an ill-designed pump, however large, cannot produce high vacuum, and most pumps act efficaciously from all clearance spaces being filled and air leaks stopped by the water passing through them; without water they would fail to make or maintain a good vacuum. Under fair working conditions, such as obtain on shipboard, a vacuum of 26 ins. can be maintained with an air-pump having a capacity of 0·3 cubic foot for each pound of condensed water passing through it. Hence for a triple engine using 16 lbs. of steam per H.P. hour at R revolutions,

$$\text{the amount passing through the pump each stroke} = \frac{16 \times \text{I.H.P.}}{60 \times \text{R}}.$$

$$\text{Then capacity of pump} = \frac{0{\cdot}3 \times 16}{60} \times \frac{\text{I.H.P.}}{\text{R}} = 0{\cdot}08 \frac{\text{I.H.P.}}{\text{R}} \text{ cub. ft.}$$

This, however, would be too small for quick formation of vacuum, and with leaky L.P. glands insufficient to keep the 26 ins. To maintain as well as form a vacuum of 28 to 29 ins. the allowance should be 0·8 cubic foot per pound of feed water. Then

Rule 132. Capacity of air-pump $= \frac{0{\cdot}8 \times f}{60} \times \frac{\text{I.H.P.}}{\text{R}}$ when worked by the engine $= 0{\cdot}0133 f \times \frac{\text{I.H.P.}}{\text{R}}$.

For triple compound engines f may be taken at 16.
,, quadruple ,, ,, 14.5.

Taking 15 as the average amount, the general rule becomes

$$\text{Capacity of air-pump} = 0{\cdot}2 \times \frac{\text{I.H.P.}}{\text{R}}.$$

It is, however, often convenient in designing and common practice to make the air-pump in proportion to the L.P. cylinder. Hence:—

Rule 132a. Capacity of Single Acting Air-Pump = 0·04 that of the L.P. cylinder.

For ferry steamers or those calling at piers, and so often stopping and required to get away at full speed rapidly, the multiplier may be 0·047 with advantage.

The common air-pump with foot, bucket, and head valves is not often used now, the *Edwards* pump with only head valves having superseded them, as being quite as efficient in every-day practice, and less costly both to make and to work by the omission of two-thirds of the valves, and less likely to get out of order also.

Weir's dual Air-Pumps when worked by an independent motor are also much in favour, especially for turbines. Here the work is done in stages, the one pump delivering semi-compressed air into the other and keeping all its air leaks sealed and its temperature low by drawing and

using some specially cooled water. The delivering pump may be warmer and not so carefully scaled.

Parsons' Vacuum Augmenter is an ingenious and simple method of getting high vacuum. In this the air-pump is placed well below the condenser and permits of a column of water to seal the air, which is drawn away by a steam ejector from the condenser bottom, and keeps the condenser itself free from air or any other gaseous matter, which would otherwise seriously reduce its efficiency.

Air-Pump Rods.—These are made of one of the rolled strong bronzes. Where a guide is fitted, and the stroke of the pump does not exceed 2¼ times its diameter, the rod (without regard to its material) may have a diameter given by,—

Rule 133. Diameter of rod $= \frac{\text{Diameter of pump}}{9} + \cdot 6$ inch.

See Table LXII. (page 163) for sizes given by this rule.

The Size of the Screw on the air-pump rod, by which it is attached to the air-pump crosshead, is given by,—

Rule 133a. Area at bottom of thread $= \frac{\text{Area of bucket}}{20 \times \sqrt{\text{diameter of bucket}}}$.

See Table LXII. (page 163) for sizes given by this rule.

This allows a stress of about 2000 lbs. per sq. inch of material in the rod for a 12-inch pump, increasing to 3800 lbs. per sq. inch in the rod for a 40-inch pump, the load on the bucket being taken as 30 lbs. per sq. inch.

When the number of reciprocations is high (say 200 to 350 per minute) the rods should be made 10 per cent. larger in diameter than given by the rule, although the stroke may be short, or even very short, since there is always a liability to sudden strain, owing to irregular or intermittent action of the pump, and the intensity of these strains will vary as the squares of the bucket speeds,—other things being equal.

Connecting-Rods for Trunk Air-Pumps.—When of the type commonly employed for driving the pumps of oscillating paddle-engines these may have the following proportions :—

Rule 134. Area of section of rod $= \cdot 01 \times$ **area of bucket ;**
or, in case of a round rod,—

Rule 134a. Diameter of rod $= \cdot 1 \times$ **diameter of bucket ;**
when two bolts are used to connect the brasses at the end,—

Rule 134b.

Diameter of each bolt (in body) $= \cdot 056 \times$ **diameter of bucket.**

Air-pump valves are usually of metal, and made as light as possible with little lift. Simple discs of sheet brass answer very well ; if fitted as tongues it is necessary to employ a metal with some spring and

capable of withstanding continuous vibratory stresses. Sheet phosphor-bronze is good.

When the air-pump valves are of india-rubber care should be taken to fit a quality that is capable of resisting the heat, &c. ; the Admiralty specification is as follows :—

"To be made of the best Para caoutchouc, with no other materials whatever than sulphur and white oxide of zinc ; the sulphur is not to exceed 1¾ per cent., and the oxide of zinc not to exceed 70 per cent. The india-rubber is to be made of the best materials, not re-manufactured, and to be of a homogeneous character throughout, thoroughly compressed, free from air-holes, pores, and all other imperfections.

"Samples of the india-rubber will be tested, and must be capable of enduring a dry heat test of 270° Fahrenheit for one hour, and a moist heat of 320° Fahrenheit for three hours, without impairing its quality."

Air-Pump Barrels.—In the mercantile marine the pump barrel is of bronze, secured to its seat and supporting the delivery chamber and hot-well by flanges. It is usual in Naval work to make barrel, head and foot boxes all one bronze casting. Whichever type of construction is used, the working barrel should be of thickness given by,—

Rule 135.

$$\text{Thickness of air-pump barrel} = \frac{\text{Diameter of pump}}{60} + \cdot 25\text{-inch.}$$

See Table LXII. (page 163) for thicknesses given by this rule.

When the pump is of the Naval type the portions above and below the working barrel may be from $\frac{1}{16}$ to $\frac{1}{8}$ inch thinner than the barrel, according to size and design.

When the bronze barrel is not enclosed in an outer casting, and there are foot valves and bucket valves fitted, it is very desirable to have a manhole in the side, through which access may be had to them without disturbing the cover or guides. Similarly, when the size of the pump renders it possible, hand or sight holes should be provided in the sides of the head box, for getting at the head valves.

The Areas through the Valve Gratings of foot, bucket, and head valves should be kept as large as possible ; on an average, those of the foot and bucket valves run about one-third of the gross area of bucket, and those of the head valve a little larger ; they can of course be increased by increasing the diameter of the pump, and decreasing the stroke.

Edwards' pumps having only "head" valves are now almost always used for air-pumps, and give every satisfaction, as they can maintain high vacuum without foot or bucket valves to trouble one, and with an unpacked bucket working with the minimum amount of friction.

Speeds of Bucket.—In ordinary mercantile engines the speed of bucket employed varies from 200 feet to 350 feet per minute ; but in Naval engines it commonly ranges from 300 feet to 450 feet, and with

pumps driven direct from the pistons is of course higher still. When the pump works in connection with a jet condenser the speed should not much exceed 200 feet per minute.

Sizes of Suction and Discharge Pipes.—For the pumps of surface condensing engines (where no jet apparatus is fitted) the diameter of suction pipe from condenser, in inches, is given by,—

Rule 136. Diamr. of air-pump suction pipe in ins. $= \frac{D}{48}\sqrt{S \times R}$,

or, **Equivalent mean speed in pipe must not exceed 2500 ft. per min.**, where D is diameter of bucket in inches, and S its stroke in feet, R the revolutions per minute.

If there is a supplementary hot-well or feed-tank the pipe connecting it with the air-pump should be of diameter given by,—

Rule 136a. Diamr. of air-pump delivery pipe in ins. $= \frac{D}{33}\sqrt{S \times R}$.

In addition, an air-pipe should be fitted as close to the pump as possible; its diameter may be,—

Rule 137. Diameter of air-pipe in ins. $= \frac{D}{100}\sqrt{S \times R}$,*

or, **Equivalent mean speed in pipe must not exceed 6400 feet per minute.**

If the pipe to feed-tank is long, or has many sharp bends in it, these points must, of course, be considered and allowed for in fixing the size.

When the pump works in connection with a jet condenser, the diameter of suction and delivery pipes may be,—

Rule 138. Diamr. of suction and delivery pipes in ins. $= \frac{D}{25}\sqrt{S \times R}$,

or, **Equivalent mean speed in pipe must not exceed 625 feet per minute.**

The delivery pipes from air-pumps of jet-condensing engines should have good large air-vessels (say capacity not less than capacity of pump) placed as near to the pumps as possible.

* *Note.*—Where there is only one delivery pipe from pump head, its diameter must be $\frac{D}{48}\sqrt{S \times R}$ ins.

Table LXII.—Air and Circulating Pumps.

Diameter of pump (single-acting).	Thickness of barrel (single-acting).	Thickness of barrel +10 per cent. (double-acting).	Diameter of rod.	Diameter of screw on outer end of rod.	Diameter of double-acting pump of same capacity (circulating only).
1	2	3	4	5	6
inches.	inch.	inch.	inches.	inches.	inches.
8	3/8	...	1 1/2	1 1/4	...
10	7/16	...	1 3/4	1 1/2	...
12	7/16	...	2	1 3/4	...
14	1/2	...	2 1/4	2	...
16	1/2	1/2	2 1/2	2 1/4	11 1/2
18	9/16	1/2	2 1/2	2 1/4	13
20	9/16	9/16	2 3/4	2 1/2	14
22	5/8	9/16	3	2 3/4	15 1/2
24	5/8	9/16	3 1/4	2 3/4	17
26	11/16	5/8	3 1/2	3	18 1/2
28	11/16	5/8	3 3/4	3	20
30	3/4	11/16	4	3 1/4	21
32	3/4	11/16	4 1/4	3 1/2	22 1/2
34	13/16	11/16	4 1/2	3 1/2	24
36	7/8	3/4	4 1/2	3 3/4	25 1/2
38	7/8	3/4	4 3/4	4	27
40	7/8	13/16	5	4	28 1/2

RECIPROCATING CIRCULATING PUMPS.

These may be double-acting, except in small ships when they are single-acting, and, as their efficiency is practically unaffected by position, they may be either vertical, horizontal, or inclined. In general design the various parts may resemble the corresponding parts of the air-pump (in practice the same patterns are sometimes used for both pumps), but circulating pumps require, in addition, large air vessels, non-return air or pet cocks, and, in the case of large double-acting pumps, a by-pass valve.

The air-vessel for a single-acting pump should have a capacity equal to twice that of the pump, when possible, but never less than one and a half.

The by-pass valve may have an area of about one-tenth the area of the bucket.

When the size of pump required would, if made single-acting, exceed 20 inches in diameter, it is better to fit a double-acting pump of half the bucket area, and thus obtain a steadier delivery, whilst at the same time reducing the magnitude of the stresses and the sizes of the various parts.

Size of Circulating Pump.—This depends on quantity of cooling water required, and number of strokes per minute.

Let Q = cubic feet of cooling water required per minute.

n = number of strokes per minute.

S = length of stroke in feet.

Then,—

Rule 139. Capacity of Circulating Pump $= \frac{Q}{n}$ cubic feet.

and,—

Rule 139a. Diameter of Circulating Pump $= 13{\cdot}55\sqrt{\frac{Q}{n \times S}}$ inches.

In determining Q, the temperature of cooling water should be taken at the highest ordinary temperature encountered on the routes the vessel is intended to steam over, *v.* Table LVIII., page 153.

Q is the quantity of cooling water in cubic feet, n the number of strokes per minute, S the stroke in feet.

If the allowance is 42 times the feed, and feed is taken at 15 lbs. per H.P., then,—

$$Q = \frac{42 \times 15}{62{\cdot}5 \times 60} \times \text{I.H.P.} = 0{\cdot}17 \times \text{I.H.P. cubic feet.}$$

Rule 140. Then Capacity of Pump $= \frac{0{\cdot}17 \times \text{I.H.P.}}{n}$ cubic feet,

if the pump is single-acting, n = Revs. per min.

Circulating Pump Rods.—For single-acting pumps, these should be of the same materials as the air-pump rods, and the same formula may be used to obtain size. See Table LXII.

Wherever possible the air and circulating pump rods should be made of exactly the same dimensions, in order that one spare rod may serve for either pump.

The rule for rods of double-acting circulating pumps will then be,—

Rule 141. Diameter of Rod $= \frac{D}{6{\cdot}4} + {\cdot}6$ inch (see Table LXII.).

Although packing is occasionally used, the buckets of circulating pumps really do not require any, and are better without it, since the serious friction is thereby saved.

The thicknesses of gun-metal barrels or liners for circulating pumps may be determined by the rule previously given for air-pump barrels (Rule 135), when the barrels are of similar type and similarly supported. When the barrel is fixed in place by means of a single flange placed at mid-length (a common method for double-acting circulating pumps), the thickness should be 10 per cent. above that given by the rule (see Table LXII.).

The areas through the foot, bucket, and delivery valves should be

kept as large as possible, but, as in the case of air-pumps, they will usually average one-third of the gross bucket area.

The bucket speeds in common use are from 200 to 400 feet per minute.

The sizes of suction and delivery valves may be as follows:—

Rule 142. Diameter of Pipe $= \frac{D}{F} \times \sqrt{S}$. ins.

Where D is the diameter of bucket in inches, S its mean speed in feet per minute, and F a co-efficient, the values of which are as follows:—

Double-acting pumps	Suction	F = 25	(mean speed in pipe,	625 feet).
	Delivery	F = 24	(,, ,,	576 ,,).
Single-acting pumps	Suction	F = 27	(,, ,,	729 ,,).
	Delivery	F = 24	(,, ,,	576 ,,).

When any pipe is under 10 inches in diameter the velocity of flow through it should be kept 10 per cent. below that allowed by the above rule. Specially long or specially tortuous pipes must also have special consideration,—the velocity of flow being reduced in accordance with the circumstances of the case.

Lloyd's rules require that a bilge injection or a bilge suction to the circulating pump shall be fitted, the diameter of which shall be at least two-thirds that of the sea-inlet.

CENTRIFUGAL CIRCULATING PUMPS.

In Naval and express work this type of pump is almost invariably used, and not infrequently fitted in other classes of mercantile steamers. It is, no doubt, preferable to the reciprocating type where large bodies of water are to be moved against a merely nominal head, and especially where the main engines run at a high number of revolutions per minute, since there is an entire absence of shock, and the work is done with a less expenditure of power. On the other hand, the centrifugal pump is more costly than the reciprocating, occupies more space, requires more attention.

Size of Pipes.—The size of pipes should be such that the speed of water does not exceed 500 feet per minute, even when the sea temperature is 75°; or, in other words, when the sea temperature is 60° the speed should be about 330 feet per minute. When the pipes are large, say over 15 inches, the velocity of flow at full speed may be higher, so that in ships of very large power it is from 600 to 700 feet.

If the pipes are not short and direct, with easy bends, the efficiency of the pump will be improved by keeping the speeds 10 to 15 per cent. per min. lower.

A convenient form of the rule for size of pipe will then be as follows:—

When the cooling water is not more than 8 lbs. per H.P. minute or about 32 times the feed, the suction and delivery pipes may be as follows:—

Rule 143.

$$\text{Diameter of cooling water pipes in ins.} = \sqrt{\frac{\text{I.H.P.}}{25}} + \tfrac{1}{2} \text{ in.}$$

For ships going to the tropics and requiring a vacuum not less than 28 inches, then

$$\text{Diameter of cooling water pipes in ins.} = 0{\cdot}28\sqrt{\text{I.H.P.}} + \tfrac{1}{2} \text{ in.}$$

Rule 143a. $\text{Diameter of pipe in inches} = {\cdot}6\sqrt{Q} + \tfrac{3}{4}$ in.,

where Q is the maximum number of cubic feet of cooling water per minute.

Size of Impeller or Wheel.—A good proportion for the diameter of pump wheel is,—

Rule 144.

$$\text{Diameter of Pump wheel in inches} = 2{\cdot}8 \times \text{Diameter of pipes,}$$

or,—

Rule 144a. $\text{Diameter of pump wheel in inches} = 1{\cdot}8\sqrt{Q}$,

where Q is maximum number of cubic feet of cooling water per minute or, reversing the equation,—

Rule 144b. $$Q = \frac{D^2}{3{\cdot}24} \text{ cubic feet.}$$

Width of Wheel at Periphery.—The width of wheel at the periphery, or width of vane at the tip may be,—

Rule 145.

$$\text{Width of Wheel at Periphery} = \frac{\text{Diameter of Pipes}}{4}.$$

Stroke of Piston.—The stroke of the centrifugal pump engine may be conveniently fixed as follows:—

Rule 146. $\text{Stroke of piston} = {\cdot}3 \times \text{Diameter of impeller.}$

Size of Steam Cylinder.—As centrifugal circulating pumps are invariably fitted with bilge suctions so as to be available as bilge pumps in case of need, and as the work done when pumping from the bilge is far in excess of that done when merely circulating water through the condenser, the size of steam cylinder must evidently be fixed with regard to the former duty.

The efficiency of the pump depends very greatly on the pipe arrangement; with direct pipes of good size, and easy bends, it may, when pumping from the bilge, approach 30 per cent., but a bad arrangement, with sharp bends, will very seriously affect it, and may

easily reduce it below 20 per cent.; in ordinary practice, where reasonable care is exercised in scheming the pipes, it may be assumed to be 25 per cent. The efficiency also tends to diminish as the lift increases, so that a rather lower efficiency should be assumed in the case of large deep ships, and *vice versâ*.

$$\text{Then } \frac{\text{Indicated Horse-power}}{\text{Water Horse-power}} = 4$$

where "water horse-power" is $\frac{\text{lbs. water pumped per min.} \times \text{lift in ft.}}{33{,}000}$

or, putting it into another shape,—

Rule 147.

Tons water pumped per hour × lift in feet × 150 = $P_m \times A \times S$,

where P_m = mean pressure in steam cylinder;
A = area of piston in square inches;
S = piston speed in feet per minute.

Speed of Periphery of Pump Wheel.—The speed of periphery of pump wheel necessary to discharge water at the velocities mentioned above against a head (h), under average conditions, is given by,—

Rule 148. $V = 11\sqrt{h}$,

where V is velocity of periphery in feet per second, and h the actual lift in feet. With a very good arrangement of pipes the co-efficient may be as low as 9, in place of 11, but a bad arrangement will send it up to 13 or over.

In ordinary circulating work when inlet and outlet are both submerged, as is usual in Navy, V requires to be about 24 feet per second (1440 feet per minute), and the resistances may therefore be assumed to be equivalent to a head of nearly 5 feet.

When pumping from the bilge, V varies from 30 feet to 50 feet per second (1800 feet to 3000 feet per minute) in ordinary Naval work, but, as shown by the above equation, it increases as the square root of the head.

Size of Bilge Suction Pipe.—A convenient rule for area of this pipe is,—

Rule 149.

One square inch area for each 7 tons per hour to be pumped.

This limits the speed of water in the suction pipe to 580 feet per minute.

It is not desirable to have the area of bilge suction more than about ·6 of the area of the circulating pipes, as the size of steam cylinder required soon becomes cumbersome for the ordinary circulating work.

The pump should be capable of performing the specified bilge duty with a steam pressure not exceeding two-thirds of the ordinary boiler pressure.

FEED PUMPS, &c.

Capacity of Feed Pumps in Jet-Condensing Engines.—This depends mainly on the degree of saltness at which the water in the boilers is to be maintained; or, in other words, on the proportion of the *gross* feed-water that is "blown off."

Let Q represent the *nett* feed-water, or the quantity required as steam by the engines, and say water in boilers is to be kept down to n times the density (or saltness) of sea-water, then the amount of sea-water that must be pumped into boilers is,—

$$\text{Gross feed-water} = \frac{n}{n-1} \times Q.$$

To reduce the time required to bring the water in the boilers up to the working level again after "blowing off," it is usual to make *each* feed pump capable of pumping *twice* this quantity; therefore,—

Rule 150. **Each pump should supply** $\frac{2n}{n-1} \times Q.$

The nett feed-water (Q) required by a jet-condensing engine, working with steam of about 30 lbs. pressure, was 26 lbs. per I.H.P. per hour.

The amount of scale or salt deposited in the boilers does not depend on the density at which the water is kept, but only on the quantity of sea-water evaporated by the boilers.

The Hydrometer, or Salinometer, by means of which the density of the water is ascertained, is, in the Navy, graduated in degrees, so that, when floating in pure water, the zero point is at the surface, and when in clean sea-water it marks 10°, when in water of twice the density of sea-water, 20°, and so on. In the Merchant Service, engineers either express the density in ounces per gallon (sea-water containing about 5 oz. per gallon), or in "thirty-twos,"—sea-water containing about $\frac{1}{32}$ of its weight of solid matter.

Capacity of Feed Pumps for Surface-condensing Engines.—It will be sufficient if the total capacity of each pump be made equal to twice the nett feed-water required; this allowance also covering any want of efficiency in the pumps.

If Q be the nett feed-water required in lbs. per hour, l length of stroke of feed pump in inches, and n number of strokes per minute; then,—

Rule 151.

Diameter of each feed-pump plunger, in inches $= \sqrt{\frac{1{\cdot}15 \times Q}{l \times n}}.$

For ordinary compound engines the nett feed-water (Q) required may be taken at 18 lbs. per I.H.P. per hour, for triple engines 16·0 lbs. per I.H.P. per hour, and for quadruple engines 14·5 lbs. per I.H.P. per hour.

Or, if the formula be written,—

Rule 152. Capacity of feed pump, in cubic inches $= \frac{\text{I.H.P.}}{\text{R}} \times \text{C}$,

the values of C will be as follows for the nett feed:—

For Compound engines, C = 8·3.
,, Triple engines, C = 7·4.
,, Quadruple engines, C = 6·7.

For actual size of pump when worked from the main engine C should be, in each case, taken at double the value to allow for wastage and sudden demands for a larger than normal supply.

Where the feed pumps are driven by the main engines, there should be, except in very small engines, two pumps, each capable of supplying the boilers at full power, and so arranged that either may be worked independently of the other, and be easily put out of gear when not required.

Relief Valves.—When the pumps are driven by the main engines, each pump should be fitted with a relief valve, of a diameter equal to two-thirds that of the delivery pipe from that pump, and loaded to 1½ times the boiler pressure. All water from relief valves should go back into suctions. Independently driven feed pumps without fly wheels need no relief valves, as they simply stop automatically and go on again when the check valves on the boilers permit.

Feed-pump valve-boxes and valves.—These should be of best bronze; and since the seats, as well as the valves, wear out rapidly, loose ones should be fitted. The faces of the valve seats should be made flat, and the seating area, or area of faces in contact, should not be less than 20 per cent. of the gross area through valve; that is,—

Rule 153.

Width of faces in contact = ·0475 × **Diameter of valve.**

The diameter of valve, measured inside the seat, should be equal to that of the delivery pipe.

The noise made by the valves on their seats may be much reduced by loading them with light spiral springs, made of plated steel or of hard brass wire, and very large ones may have a cataract buffer.

Feed pipes.—The pipes leading to and from the feed pumps should be of such size that the velocity of flow, when working steadily at full power, does not exceed 500 feet per minute. The

velocity should also be less as the pipes get smaller, to allow for the increased percentage of friction; and, if 400 feet be adopted as the limit in a 4-inch pipe, the limit in a 1½-inch pipe should be 250 feet; or, more briefly, velocity for any diameter of pipe should not exceed that given by,—

Rule 154. $V = 200\sqrt{\text{Diameter}}$,

where V is in feet per minute and diameter is in inches.

The maximum deliveries of pipes, on ordinary service, may be shown in tabular form as follows:—

Table LXIII.—Delivery of Feed Pipes at Service Speeds.

Diameter of pipe.	Delivery in cubic feet per minute. (C)	Delivery in lbs. per hour.	Diameter of pipe.	Delivery in cubic feet per minute. (C)	Delivery in lbs. per hour.
1½	2·45	9,260	3½	20·43	77,200
1¾	3·60	13,600	3¾	24·30	91,800
2	5·04	19,050	4	28·63	108,200
2¼	6·79	25,660	4¼	33·30	125,800
2½	8·81	33,300	4½	38·42	145,200
2¾	11·22	42,400	5	50·00	189,000
3	13·94	52,700	5½	63·36	239,500
3¼	17·00	64,200	6	78·92	298,300

Then, to find diameter of feed pipe, determine value of C in equation,—

Rule 155. $$\frac{D^2 \times S}{184} = C,$$

where D is diameter of plunger in inches, and S its mean speed in feet per minute; look for the corresponding figure in column 2 or 5 of Table; and opposite, in column 1 or 4, will be found the appropriate diameter of pipe. If the exact figure does not appear in column 2 or 5, take the next higher. Of course, if the pump is single-acting, the deliveries will be one-half those given in above Table.

The following is a useful rule for determining the size of feed pipes of triple and quadruple engines:—

Rule 156. **Diameter feed pipe** $= \frac{\sqrt{\text{I.H.P.}}}{K} + 0{\cdot}5$ inch.

For engines having their own feed pumps K = 18; when fed by independently worked pumps K may be 22.

Feed-Pump Plungers.—These should be of good bronze, and when of the single-acting ram type may be of thickness given by,—

Rule 157.

$$\text{Thickness of metal of plunger} = \sqrt{\frac{P \times D}{4660}} + \cdot 15 \text{ inch},$$

where P is working pressure in lbs. per square inch, and D is diameter of ram in inches.

Feed-Tanks.—To avoid waste of water through the occasional overflow of the hot-well, it is the rule in the Navy to fit a feed-tank, or supplementary hot-well, into which the air pumps deliver, and from which the feed pumps draw; and, where fresh water only is used, such an arrangement is quite necessary.

The capacity of the tank is usually about 1 cubic foot for every 20 I.H.P. The tanks are commonly of $\frac{5}{16}$-inch steel plates, flanged over and single riveted, stayed for a pressure of 10 to 15 lbs. per square inch, and galvanised after completion. Each tank should have a manhole, an air-pipe, an overflow pipe, three or four zinc blocks to prevent corrosion, and a water-gauge giving a visible range of at least 3 feet; and where there are two or more engine-rooms, with a tank in each, the various tanks should be connected one with another. These tanks are frequently built into the structure of the ship. The feed-tanks should be placed as close to the air-pumps and as low down as possible; where they are under the platforms a float and index is more convenient than a water-gauge.

Reserve Tanks, &c.—Where the boilers are fed with fresh water only, a sufficient reserve supply should be carried to provide about 21 cubic feet, or say 1300 lbs. "make-up," or auxiliary feed, per 100 I.H.P. per 24 hours.* This supply may either be carried in reserve tanks, in the double bottom or elsewhere; or may be produced as required by evaporators delivering into the L.P. valve-casing, or into the condenser; or, a combination of these two methods may be employed. In recent Naval practice the tanks have a capacity of about 1 c. ft. for every 6 I.H.P.

Feed Heaters.—Various types of feed heater are now in common use; and, where space and weight allow of their adoption, they are, without doubt, a very desirable adjunct, since they keep a great deal of deposit out of the boilers, and,—when the exhaust steam from the auxiliary engines can be used in them,—save a considerable amount of heat that would otherwise be wasted.

Board of Trade Rules relating to Feed Pumps, Pipes, &c.

Paragraph 136.—Each boiler of a passenger vessel, whether old or new, should be fitted with suitable check valves between it and the feed pipes, and the boilers of all *new* passenger vessels, should be

* With care the *average* consumption may be kept down to about one-half this quantity.

fitted with separate feeding arrangements in addition to, but unconnected with, the main feed pipes and valves. It is desirable that the main feed check valve chest on each boiler should be separate and distinct from that of the auxiliary feed, and that a stop cock or stop valve should be fitted in each chest or between each chest and the boiler, so that the latter may be shut off, and either of the check valves examined while the other feed is at work. In very small vessels an efficient hand-pump, instead of the usual donkey pump, may be passed if the Surveyor has satisfied himself as to its efficiency when steam is up, and provided that there are separate feed pipes and valves as directed above.

The Surveyor should discourage the practice of using the same pump for the bilges and feeding boilers.

Lloyd's Rules relating to Feed Pumps, Pipes, &c.

Sec. 7. The engines are to be fitted with two feed pumps, each capable of supplying the boilers; the pumps, &c., to be so arranged that either can be overhauled whilst the other is at work.

3. The main feed pumps may be worked by independent engines, provided they are fitted with automatic regulators for controlling their speed. If only one such pump is fitted for the main feed, the auxiliary feed pump required by paragraph 25 should also be fitted with an automatic speed regulator.

6. A steam pump is to be provided capable of supplying the boilers with water, this pump to be provided with suctions from hot-well and from sea. A steam pump is to be so fitted as to pump from each compartment, to deliver water on deck, and, if no hand pump is fitted in engine-room, it must be fitted to be worked by hand. In small vessels in which only one steam pump is fitted, it must comply with all these requirements.

BILGE PUMPS.

There is no definite basis of calculation for the size of these pumps, and no generally recognised rule. For jet-condensing engines they were generally of the same size as the feed pumps, but for surface-condensing engines the following rule may be used:—

Rule 158. Capacity of Bilge Pump $= \frac{\text{Capacity of L.P. Cylinder}}{350}$.

The following rule is, however, more appropriate to the requirements, D being the displacement of the ship in tons; there may be from one to four pumps.

Rule 158a.—Total Bilge Pumps capacity in cubic inches should be not less than $3 \cdot 5 \times D^{\frac{2}{3}}$.

Where the pumps are driven by the main engines there should be,—except in the case of very small ships,—two pumps of the size given

by these rules, arranged so that either may be worked independently of the other, and readily put out of action when not required.

Covers of valve-boxes and mud-boxes should be secured by hinged bolts and wing nuts, in order that they may be easily and quickly removed, and replaced whenever necessary.

All strainer, or mud, boxes should be placed in easily accessible positions *above* the floor plates, as terminal rose-boxes are always troublesome, and may be a cause of serious danger.

The "directing" or "distribution" valve-boxes should also be above the floor plates, and easily accessible; and each cover or hand-wheel should bear a name plate indicating the compartment with which it opens communication; a very good, and also very cheap, plan is to cast the required letters or words on the upper face of the hand-wheel rim.

Where one bilge pump must also act as wash-deck or fire pump (to comply with Board of Trade regulations), a three-way open-bottom cock, with one port in plug, should be fitted, to prevent the possibility of sea-water entering the ship; when the pipes are over three inches in diameter, however, it is not always convenient to fit a cock, and non-return valves may be fitted,—though they are less safe than the cock.

Board of Trade Regulation referring to Bilge Pipes, &c.

82. **Sounding Pipes.**—Sounding pipes should be fitted from the upper deck for ascertaining the depth of water in each compartment.

The sounding pipes for the ballast tanks under the engine and boiler room and under the tunnel floor may, however, be short, provided that they are fitted with screwed caps at the upper ends, or with cocks having the handles secured to the plugs.

83. **Engine-room Pump Fittings.**—Suction pipes connected with pumps worked by the main and donkey engines should be carried through the bulkheads into all the compartments fore and aft of the engine-room except the compartment in front of the collision bulkhead, so that each compartment can be pumped out separately by the engines. When a double bottom is fitted extending the full length of a compartment, with waterways on each side, a bilge suction should be fitted to each waterway. This requirement need not, however, be enforced in small compartments near the ends of the vessel. The pipes should be well secured where they pass through the bulkheads. In all new steamships, the cocks or valves which are fitted for the purpose of shutting off or controlling the flow of water through these pipes should, unless situated in the compartment occupied by the pump to which the pipes are connected, be provided with means by which they may be manipulated from a height well above the deep load line, preferably the upper deck.

The arrangement of bilge suction pipes should be such that water cannot pass through them from one compartment to another. Valves

in bilge distribution boxes should be of the non-return type, and if cocks are used instead of valves, suitable non-return valves should be fitted in the pipes to prevent water passing from one compartment to another in the event of the cocks being left open.

The free end of each hold suction pipe should be fitted with a suitable rose-box or strum, and the tunnel-well suction pipe should be similarly fitted, unless it is provided with a mud-box of the form required for the pipes in connection with the engine-room and stokehold bilges.

Each main and donkey-engine suction pipe in connection with the engine-room and stokehold bilges should be provided with an efficient mud-box, or other similar appliance, placed above the platform, or in any other position in which it will be accessible at all times. The pipe leading from each mud-box to the bilge should, when practicable, be straight, and the cover of the mud-box should be secured in such a manner as to permit of it being expeditiously opened or closed.

The connection from the donkey engine to the bilge main should be by means of a switch cock or non-return valve.

An efficient bilge injection should be fitted to each main circulating pump.

Lloyd's Rules relating to Bilge Pumps, Pipes, &c.

2. The engines are to be fitted with two bilge pumps, which are to be so arranged that either can be overhauled whilst the other is at work.

4. A bilge injection, or bilge suction to the circulating pump, is is be fitted.

5. The engine bilge pumps are to be fitted capable of pumping from each compartment of the vessel, the peaks excepted. All bilge suction pipes are to be fitted with strum-boxes or strainers, so constructed that they can be cleared without breaking the joints of the suction pipes. The total area of the perforations in the strainers should be not less than double that of the cross section of suction pipe. The mud-boxes and roses in engine-room are to be placed where they are easily accessible, and to the satisfaction of the Surveyor.

6. A steam pump is to be provided capable of supplying the boilers with water, and is to have suctions from hot-well and sea. A steam pump is to be so fitted as to pump from each compartment, to deliver water on deck, and, if no hand pump is fitted in engine-room, it must be fitted to be worked by hand. In small vessels in which only one steam pump is fitted, it must comply with all these requirements.

10. Bilge suction pipes to be arranged to pump direct from each compartment,—the roses to be fixed in easily accessible places; for numbers and sizes of suctions required, *see* Lloyd's Rules relating to holds of ships, p. 308.

Sec. 2.—5. Cocks and valves connecting all suction pipes to be fixed above the stokehole and engine-room platforms.

6. The arrangement of pumps, bilge injections, suction and delivery

pipes is to be such as will not permit of water being run from the sea into the vessel by an act of carelessness or neglect. Any defective arrangement to be reported to the Committee.

See also **Lloyd's Requirements** for sea connections, page 265.

PUMP LEVERS AND LINKS, &c.

In order to obtain quantitative results for guidance in proportioning pump gear, it is necessary to assume some maximum load per square inch of bucket, and the load here assumed is 30 lbs. for both air and circulating pumps.

Pump levers.—The strength of these may be determined by means of the ordinary rule for beams,—

Rule 159. $$W = \frac{fz}{l},$$

and,—when the central hole cut in the plate to admit the gudgeon boss does not exceed ·4 × depth of beam, and the attachment of the central boss or gudgeon flange is made with rivets in the usual way,—the values of f may be as follows :—

For thin plates, $f = 4500$ for screw engines, and 5500 for paddle engines.

For solid levers (forged) $f = 5500$ for screw engines, and 6500 for paddle engines.

In determining $z \left(= \frac{bd^2}{6} \right)$, b is four times the thickness of the plate, for ordinary plate beams, and d is the total depth or width of the plate at the gudgeon.

A lower value of f is adopted for screw engines because, by their higher speeds, there is the liability to severe strains from sudden reversals and loads of the air-pump. When air and circulating pump are both driven by the same pair of levers, the values of f should be reduced 15 per cent,—as one lever of the pair may have to carry considerably more than half the load at times.

Pump links.—For the various reasons stated above, the bolts in pump links require to be made very heavy.

Rule 160.—The link bolts of paddle engines, when two in number, may carry loads equal to those given in Table XLIII. ; and when four in number, 10 per cent. less, or 20 per cent. less according as air-pump only, or both air and circulating pump, are driven by one pair of levers. The link bolts of screw engines (usually four in number) may, in similar circumstances, carry loads 30 per cent., and 40 per cent below those given in Table XLIII.

The loads that link bolts may carry under the various conditions mentioned will then be as follows:—

Table LXIV.—Strengths of Pump Links.

Diameter of link bolt in inches.	Total load in lbs. on one bolt.				
	Paddle Engine.			Screw Engine.	
	Two bolts.	Four bolts (air-pump).	Four bolts (air and circulating pumps).	Four bolts (air-pump).	Four bolts (air and circulating pumps).
1	2,150	1,930	1,720	1,500	1,300
1⅛	3,000	2,700	2,400	2,100	1,800
1¼	4,200	3,800	3,400	2,900	2,500
1⅜	5,400	4,800	4,300	3,800	3,200
1½	7,100	6,400	5,700	5,000	4,200
1⅝	8,500	7,600	6,800	6,000	5,100
1¾	11,000	10,900	9,300	7,700	6,600
1⅞	13,100	11,800	10,500	9,100	7,800
2	16,100	14,500	12,800	11,200	9,600
2¼	20,400	18,300	16,200	14,200	12,200

When the links are composed of pairs of parallel bolts, on to the ends of which the brasses are threaded, it will generally suffice to make the diameters of the middle portions of the bolts, between the brasses, the same as those of the end or bolt portions,—since the compressive stresses are usually much less than the tensile.

When the length of the plain or middle portion of the bolt reaches 25 diameters, however, it is advisable to substitute one central rod with T ends for the two parallel ones.

If, in any case, the links are so placed that the compressive stresses exceed the tensile, and if the length of link be more than 10 diameters of bolt or pillar, their size had better be determined by the rules for struts fixed at both ends.

As the stroke of the pumps is commonly less than the stroke of the piston, the load on the links at the piston-rod end of the levers is less than the load on those at the pump end in inverse proportion to their distances from the fulcrum of the lever.

Surfaces of pins.—Rule 160a.—The surfaces of the various pins (diameter × length) to which the pump links are attached, should be such that the load does not exceed $\frac{18,000}{\sqrt{Rd}}$ lbs. per square inch, R being the revolutions and d the diameter of pin; generally the working load should be 70 per cent. of this maximum.

Pump crossheads.—These should be calculated as beams loaded in accordance with the circumstances of the case.

Considering the sections usually employed, and the nature of the

load, the value of f for mild steel may be taken at 8600. For further guidance see general formulæ for beams.

Pump lever gudgeons.—The load on the pump lever gudgeon, and on its bearings, cap bolts, &c., is the sum of the loads on the two ends. The cap bolts should not carry greater loads than link bolts of corresponding diameters. (*See* Table LXIV., page 176.)

The bearings may carry a load not exceeding $\frac{16{,}000}{\sqrt{Rd}}$ lbs. per square inch. The diameter of gudgeon will be most conveniently determined by considering the stresses to which it is subject as shearing stresses simply; then,—

Rule 161. **Diameter of Gudgeon** $= \sqrt{\frac{S}{K}}$

where S = the sum of the total loads on the two ends of the pair of levers;

K = 1200,—when there are two bearings, one close to each lever, as is usual in screw engines;

K = 1600,—when there is only one lever in place of a pair, with very short gudgeon and bearing at each side close to lever, as frequently fitted in paddle engines.

SLIDE VALVES.

Travel of Valve.—**Rule 162.**—The travel of a slide-valve should be proportional to the length of the steam-port in cylinder face, measured parallel to axis of slide-rod, and not less than twice the amount.

As the work done in moving the valve is nearly proportional to the length of travel, it is important to keep this latter as short as possible, and the best way of effecting this is to use double- or triple-ported valves.

Single-ported valves should not be used for cylinders of more than 25 inches diameter; whilst triple-ported valves can only be conveniently employed for large engines of fairly long stroke, and are therefore only applicable in Naval work when the piston speed is very high and the cylinders large.

Since the introduction of the triple engine, piston valves have been reintroduced, and are always used for H.P. cylinders, and generally for the M.P. also; their worst feature is their liability to stick when any dirt or grit is deposited on them, and the consequent severe strains that may be thrown on the valve-gear; also, as ordinarily fitted, they are not so steam-tight as flat valves.

Surface of Valve.—Flat slide-valves sometimes give unsatisfactory results from want of sufficient bearing surface,—so that the pressure per square inch of rubbing surface is so great that undue wear, cutting, &c., result.

Relief-rings.—Flat slide-valves should be fitted with some arrangement for relieving the pressure on the back, to reduce wear and tear, and save the driving power; they also require a spring of some sort to replace them against the face, after they have been forced from it by the presence of water in the cylinder, or by other causes.

Rule 163.—The pressure per square inch of rubbing surface, imposed by the springs, should not exceed 2½ lbs.

Small valves have one relief-ring, and very wide ones two rings side by side; the areas may be made as large as can be got on the valve backs, when the relief pipe from M.P. valve is led to L.P. receiver, and that from L.P. valve to condenser in the usual manner; a greater difference of pressure than given by this arrangement is not desirable, and any leakage will be a serious loss.

Port openings.—With given leads and cuts-off, both reckoned as percentages of the stroke of piston, the openings to steam are proportional to the travels, and may therefore be increased or diminished by increasing or diminishing the travel.

Lead of valve.—The lead given to the valve is generally decided arbitrarily, and varies from a bare $\frac{1}{32}$ inch in small auxiliary engines to 1¼ inch in large L.P. valves; but the piston speed and momentum of moving parts should be taken into consideration, as in M.P. and L.P. cylinders the compression is rarely sufficient in itself to absorb all the momentum, and a certain amount of lead is necessary to prevent shock.

The extent to which the engine will probably be run "notched" or "linked" up should also be considered.

Ordinarily, when the valves are properly set, the lead at the back or top end will be about half that at the front or bottom end, and the cut-off at the crank end will be earlier than at the other, but the cut-off may be the same at each end if the leads are different. With a valve cutting off late, inside lap is necessary to give the compression sufficient for quiet working; but it is a means of throttling, as well as delaying the period of exhaust. With an early cut-off sufficient compression can be obtained and a freer exhaust enjoyed if there is negative lap.

For a quick-running compound or triple engine, exhaust from H.P. cylinder should commence at ·85 of the stroke, and for a slow-working paddle engine, not later than ·95 of the stroke.

Proportions of slide-valves.—Figures 26 and 27 show some of the elementary proportions of the common and of the Trick valve.

Let x be the *outside* lap of the valve at the front end, and y that at the back end. Then (Fig. 26),—

$$H = \frac{G}{2} + x\text{; and } K = \frac{G}{2} + y.$$

Also, let z be the *inside* lap at the front, and w that at the back. Then (Fig. 26),—

$$B = \frac{F}{2} - z\text{; and } C = \frac{F}{2} - w.$$

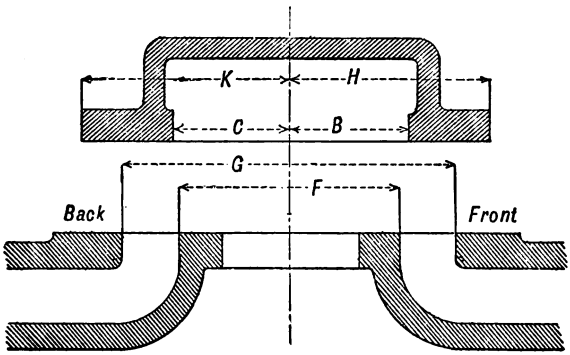


FIG. 26.—Proportions of a Common Valve.

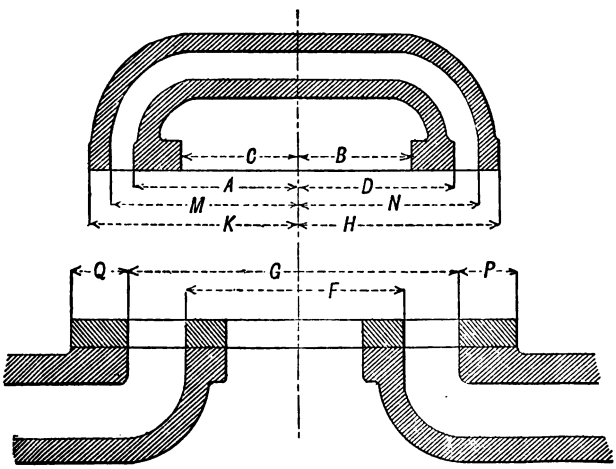


FIG. 27.—Proportions of a Trick Valve.

Referring now to Fig. 27,—let x, y, z, and w be the laps as before. Then,—

$$H=\frac{G}{2}+x\text{; and } K=\frac{G}{2}+y.$$

Also,—

$$B=\frac{F}{2}-z\text{; and } C=\frac{F}{2}-w.$$

$$A=\frac{G}{2}+\tfrac{1}{8}\text{ inch; and } D=\frac{G}{2}+\tfrac{1}{8}\text{ inch.}$$

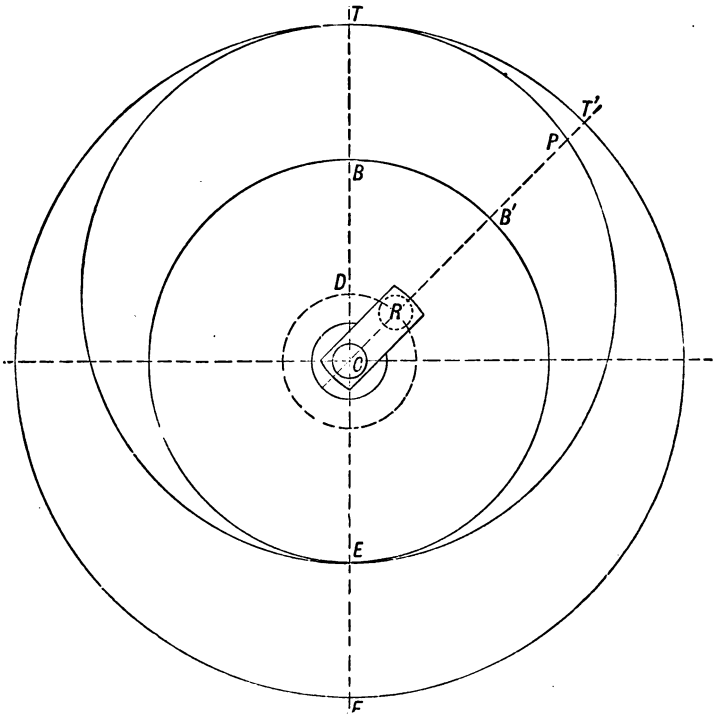


FIG. 28.—Diagram of the Piston Path.

The openings through the valve laps or covers must be as large as possible, but need not exceed the ordinary opening of the valve to steam at the outer edge; then,—

$$G + P = K + N; \text{ and } G + Q = H + M.$$

Valve diagrams.—Figures 28 and 29, when combined and used as described below, show at a glance the complete cycle of the operation of any proposed valve, and also the effect of varying any one of the elements,—travel, laps, leads, or openings.

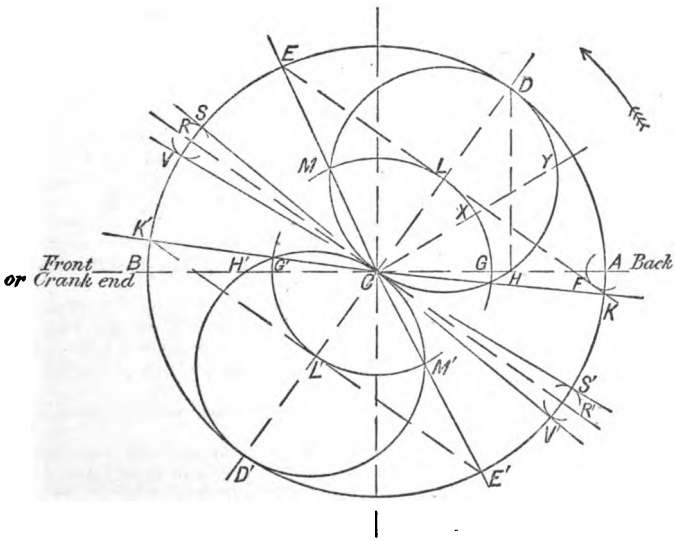


FIG. 29.—Zeuner's Diagram for the Common Valve Motion.

In constructing the diagram for a proposed valve, the diagram Fig. 29, the extreme diameter of which is equal to the travel of the valve, should be placed in the centre of the diagram Fig. 28,—the former being drawn full size, and the latter ¾ inch or 1 inch to the foot, as found most convenient.

Referring to Fig. 28, the outer circle TT'F represents the top or back end of the cylinder; the inner circle BB'E represents the bottom, front, or crank end; and the eccentric circle gives the position of the piston corresponding to any angular position of crank. The diagram is drawn by setting off CD equal to crank, and DT equal to connecting-rod, and then swinging DT round on D as a centre.

T is the position of piston on top or back centre, and E its position on bottom or front centre; and the position corresponding to any other position of crank,—such as CR,—is found by producing CR to cut the circles, when PT′ is the distance of the piston from one end of cylinder, and PB′ its distance from the other.

Referring now to Fig. 29, the problem most frequently met with is, —Given the travel, leads, and cuts-off, to determine the laps and position of eccentric,—and it is solved as follows:—

Draw CE so that, when produced, it will cut the eccentric circle at a point corresponding to the given point of cut-off (*i.e.*, if cut-off is to be at x inches from the back end of the stroke, the point in the eccentric circle must be x inches from the outer circle); then ACE is the angle through which the crank must move to arrive at the position of cut-off. With A as centre, and radius AF equal to the lead, draw part of a circle, and then draw EK. Next, draw CD perpendicular to, and bisecting EK in L. Then CL is the lap required, and BCD is the angle between the crank and eccentric; and since CD is the half-travel, LD is the maximum opening of port.

To extend the usefulness of the diagram,—on CD as diameter, describe a circle; and from centre C with radius CL, strike the arc GLM, which is called the lap circle. The part GH is then equal to AF, and represents the lead, or opening of valve when the crank is "on the centre," or in the position CA. XY likewise represents the port opening when the crank is in position CY.

To determine the operation of the valve at the other end, produce DC to D′, and describe a circle on CD′. Then let H′G′ be the lead at this end; from C, with radius CG′, draw the lap circle G′L′M′, and through C and M′ draw CE′. Then CE′ is the position of the crank at cut-off, and CG′ is the lap.

The positions of the crank, when the port commences to open, are CK and CK′.

If the valve has no inside lap (*i.e.* if the ports are both just closed to exhaust when the valve is in mid-position), release at one end and compression at the other will commence simultaneously when the crank reaches the positions CR and CR′, which are at right angles to DD′.

If, however, the valve has positive inside lap at the front end, and negative inside lap of equal amount at the back end, the positions of release and commencement of compression will be altered as indicated, back release and front compression occurring at S, and front release and back compression at S′; or, if laps be reversed, at V and V′.

The points S, S′ and V, V′ are obtained by striking the small arcs indicated with the inside lap as radius; of course, if the lap is positive, the valve will open for exhaust later and close earlier; but if the lap is negative, the exhaust will be earlier and the compression commence later.

To obtain the position of piston corresponding with the various crank positions, each radial line indicating a crank position must be continued until it meets the outer circle TT′F, and the position of piston can then be scaled off.

Effect of linking up.—The effect of "linking up" on the operation of the valve is most readily exhibited by the application to the above diagrams of the following very closely approximate construction, suggested by the late Mr Macfarlane Gray:—

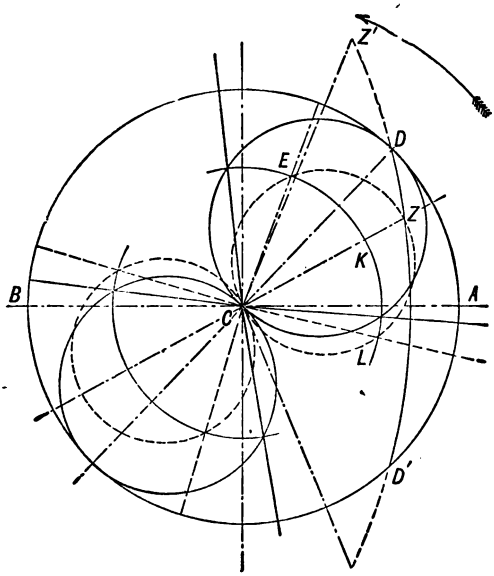


FIG. 30.—Diagram showing the effect of "Notching up."

Suppose the link (Fig. 31) to be notched up so that the link block M is distant MT from the point at which the eccentric rod is attached to the link.

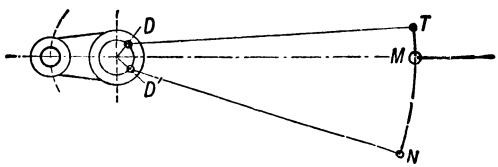


FIG. 31.—Link Motion "Notched up."

Draw the valve diagram (Fig. 30) due to the position and throw of the eccentrics in Fig. 31, and through D and D′ draw the arc of a circle with a radius found as follows:—

Rule 164. $$\text{Radius} = \frac{DT \times DD'}{2TN} \text{ (see Fig. 31).}$$

Then divide this arc at Z so that DZ is to DD′ as TM is to TN; join CZ, and on it as diameter describe a circle cutting the lap circle in L and E; and draw the lines CL and CE, the former indicating position of crank when valve opens, and the latter the position when valve closes.

It will thus be seen that the effect on the valve of notching the link up to the point M is the same as though it were driven by an eccentric having the angular position and eccentricity CZ; the lead is earlier, the opening is reduced to KZ, the cut-off is earlier, and a further examination would show that release occurs earlier, and compression commences earlier than when in full gear.

Open and crossed rods.—When the eccentric rods are arranged as shown in Fig. 31, they are said to be "open," but when the gear is so arranged that D is joined to N, and D′ to T, whilst the crank remains turned away from the link TN, the rods are said to be "crossed"; in the diagram for this latter case the arc DD′ must be drawn convex towards C, or its centre must be on side A.

When it is intended to work the engines linked up to any considerable extent, the rods should be of the "open" type, as a greater range of expansion can be obtained with less reduction of port openings than is possible with "crossed" rods.

Overhung gear.—Where the valve gear is of the "overhung" type (*i.e.* the type in which T is always outside of M, even in full gear), the diagrams are constructed on the same principle as Fig. 30, and the position and eccentricity of the eccentrics are determined by producing the arc DD′, and taking a point Z′ *beyond* D, so that Z′D is to (DD′+2Z′D) as TM is to TN, and joining CZ′. CZ′ is the required eccentricity of the sheaves, and BCZ′ is the angle between them and the crank.

Obliquity of eccentric rods.—The valve diagram, as above described, takes no account of the effect produced by the obliquity of the eccentric rods; when the rods are so short as to make this disturbing effect noticeable, the diagram may be corrected as follows:—

From A and B (Fig. 32) drop perpendiculars AN and BN′ upon RR′, and through N and N′, with length of eccentric rod as radius, strike an arc; also, with same radius, strike similar arcs through O and P and O′ and P′. Then, if the points where these arcs cut the travel circle be joined with C, the corrected crank positions, at which the various events of the cycle occur, will be obtained; the radial lines indicating these new crank positions may then be produced

outwards so as to cut the eccentric circle and indicate the piston positions as before. The correcting arcs for the exhaust laps have been omitted as tending to complicate the diagram, but they will be drawn on either side of the arc NN′, at distances from it equal to the laps, and will give new points on the travel circle in a precisely similar way.

When the eccentric rods are very short, and the locomotive or "slot" link is used, as in some types of horizontal engine, no diagram

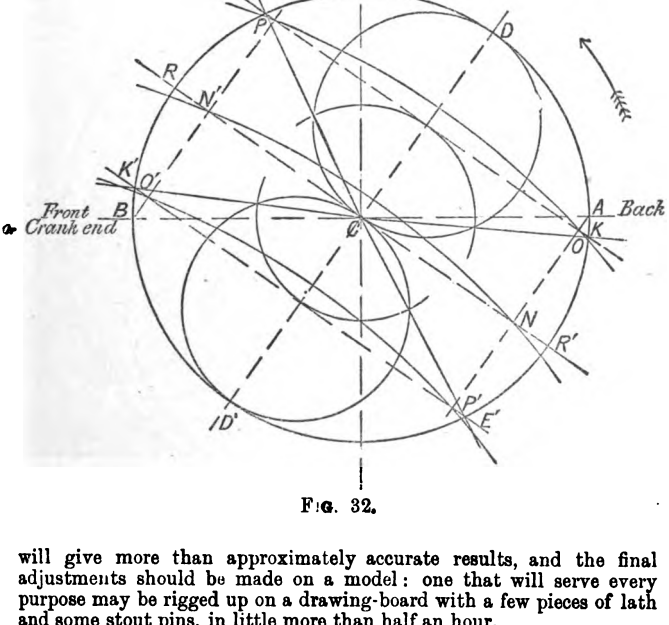


FIG. 32.

will give more than approximately accurate results, and the final adjustments should be made on a model: one that will serve every purpose may be rigged up on a drawing-board with a few pieces of lath and some stout pins, in little more than half an hour.

For the solutions of other problems connected with the diagram for the ordinary slide-valve, and for the methods of constructing diagrams for various types of expansion valve, see Mr Seaton's "Manual of Marine Engineering."

Diagram for oscillating engine.—When the diagram Fig. 28 is applied to the case of an oscillating engine, the eccentric circle deviates slightly from a true circle, owing to the fact that there is

virtually a connecting-rod of varying length. The necessary correction in the eccentric circle may be made as follows :—

Let A be the trunnion centre, and BLE the crank path ; divide the half-crank path into any number of equal parts (say six), at K, H, G, &c. ; join each point with A ; and from A, with radius AB, strike the arc BDC. Then, referring to Fig. 34, set off the corresponding points E, F, G, &c., and transfer the lengths ED, FD, GD, &c., from Fig. 33 to the positions indicated by the corresponding letters on Fig. 34, and through the points D, D, D, &c., draw a curve ; this curve will

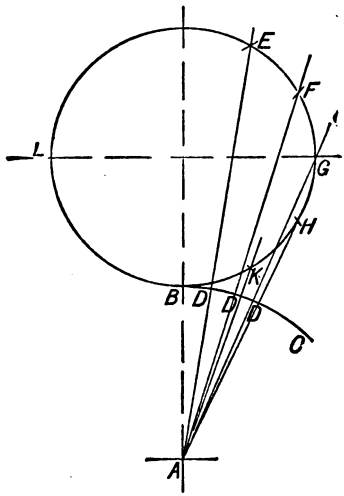


FIG. 33.

then be the true curve of successive positions of piston corresponding to the crank positions indicated by any radial lines.

GL in Fig. 34 (p. 187) must, of course, represent GL in Fig. 33, either to the same or some other convenient scale.

VALVE GEAR.

It must be understood that the various proportions given in the following sections on "Valve Gear" and "Eccentrics" are such as are suitable for *ordinary* speeds of engines of the two categories, Naval and Mercantile, and that both increased strengths and increased surfaces will be required if the engines are to run at exceptionally high numbers of revolutions.

Valve or slide rods.—The power requisite to move a slide-valve on its face is a constantly varying quantity, and care must be taken to use its maximum value in all calculations for dimensions of valve gear.

To obtain this, the reduction due to any relief arrangement, or to the pressures on some portions being in equilibrium, should be neglected,

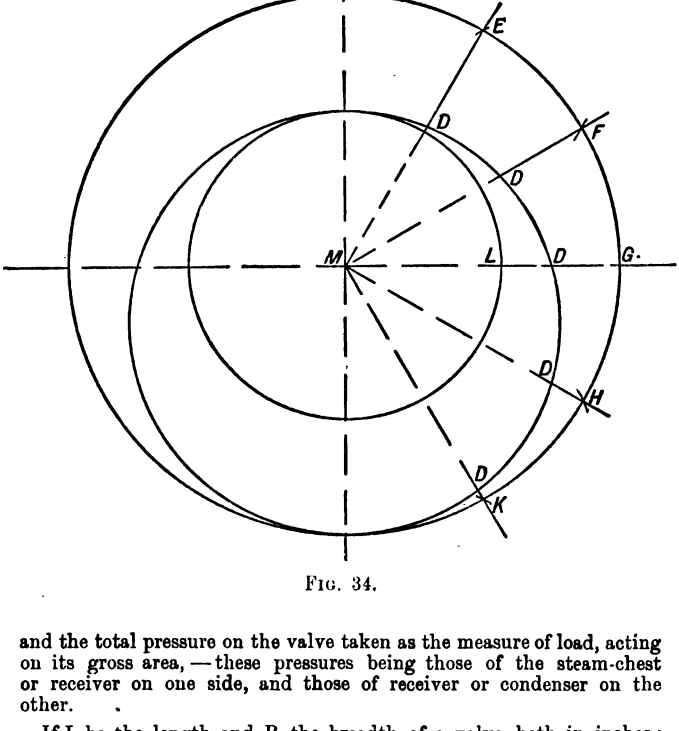


FIG. 34.

and the total pressure on the valve taken as the measure of load, acting on its gross area, — these pressures being those of the steam-chest or receiver on one side, and those of receiver or condenser on the other.

If L be the length and B the breadth of a valve, both in inches; p the maximum pressure per square inch, obtained as described above, and ·2 the co-efficient of friction for metallic surfaces rubbing together dry; then,—

Rule 165. Load on valve-rod $= \cdot 2\,(L \times B \times p)$ lbs. ;

and,—**Rule 166,—**

Smallest diameter of valve-rod $= \sqrt{\frac{L \times B \times p}{12500}} + \cdot 25$;

and,—**Rule 167,—**

Diameter of valve-rod in gland $= \sqrt{\frac{L \times B \times p}{6500}} + \cdot 25.$

The value of p should be taken as 30 lbs. for L.P. valves, and, as it is desirable to have the rods all of the same dimensions, the highest value of the above expression for any one cylinder should be used for all.

When piston-valves are used for the H.P. cylinders, the valve-rod is usually made of the same size as those for the flat slides of the M.P. and L.P. cylinders.

Theoretically, the power required to move a piston-valve should be little more than is necessary to overcome the inertia and friction at the stuffing-box, but practically,—as mentioned under "Slide-valves,"—the power required is in excess of this, and may be largely so.

The diameter of rods for piston-valves may be determined by the following:—

Rule 168. Diameter rod $= \frac{\text{Diameter valve} \times \sqrt{p}}{F} + \cdot 6$ in.

p for H.P. cylinders is the boiler-pressure.
p ,, M.P. ,, ,, half ,,
p ,, L.P. ,, ,, 30 lbs. ,,

For very light engines, as in destroyers, $F = 100$.
For express steamers, short service, and Naval ships generally, $F = 90$.
For merchant ships generally, $F = 84$.

In practice the rod of the H.P. valve is about $\frac{1}{8}$th the diameter of the valve ; that of the M.P. about $\frac{1}{8}$th ; and that of L.P. cylinder $\frac{1}{10}$th.

For convenience the valve-rods are of one size throughout.

Rule 169. The cap-bolts for valve-rod end or eye must each carry a load equal to $\frac{\cdot 2\,(L \times B \times p)}{2}$, and the size suitable for this load will be found from Table XLIII., page 103. Their diameter will be found to very closely approach $\cdot 5D + \cdot 2$ inch, where D is diameter of valve-rod found by Rule 166 above.

Valve-rod guides.—All valve-rods should be fitted with guides, placed as close to the link as possible, and, if not fitted with balance pistons, they should have guides at the tail ends as well.

The most satisfactory form of guide is one resembling a piston-rod slipper guide of the "single" type (*i.e.* one shoe and one guide for both ahead and astern) in miniature,—the shoe being cast with a

cylindrical crosshead which is bored to fit the valve-rod, and secured on it by a through cotter.

When this type is not adopted, and the guide is made to embrace the valve-rod just below, or outside of, the gland, a square or two flats should (whenever space is available) be formed on the rod to take the guide.

Reversing quadrants, or "links."—The old slot-link still retains its place, as the most convenient of application in oscillating paddle engines, and in some types of screw engines. For small fast-running engines it is quite the safest link and still retained by locomotive engineers.

Its principal proportions should be as follows—the unit D being the diameter of valve-rod as found by Rule 166.

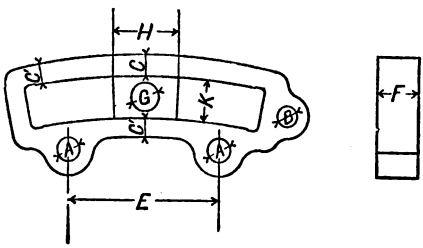


FIG 35.

Rule 170, a to k.

(*a*)	Centres of eccentric-rod pins (E)		2·75 to 3 × throw of eccentrics.
(*b*)	Breadth of link (F)		1·1 D + ·4 inch.
(*c*)	Thickness of bar (C)	with bridge	D + ·25 ,,
		without ,,	1·1 D + ·25 ,,
(*d*)	,, ,, (C′)	with ,,	·9 D + ·25 ,,
		without ,,	D + ·25 ,,
(*e*)	Length of block (H)		2·5 D + 1 ,,
(*f*)	Width of slot (K)		1·4 G + ·35 ,,
(*g*)	Diam. of eccentric-rod pin (A)	solid bush	D + ·25 ,,
		adjustable brasses	1·1 D + ·25 ,,
(*h*)	, drag-rod pin (B)	as in fig.	·55 D + ·25 ,,
		when on bridge	·8 D + ·25 ,,
(*j*)	,, block-pin (G)	when overhung	1·4 D + ·25 ,,
		both ends secured	1·3 D + ·25 ,,
(*k*)	,, drag-rod at ends	if one	·7 D + ·25 ,,
		if two	·48 D + ·25 ,,

Theoretically, surfaces may be a little smaller for paddle engines, but it is better to give the extra surface and make them same as for screw engines.

For all vertical engines, and in all other cases where possible, excepting where L.P. cylinder is below, say, 30 inches in diameter, the double-bar link is preferable, as large working surfaces and pins are readily obtained, and they are more accessible and easier to adjust.

The principal proportions of the double-bar link should be as follows, —the unit being again D, or diameter of valve-rod as found by Rule 166.

Rule 171, a to n.

(*a*) Centres of eccentric-rod pins . . .	3 × throw of eccentrics, or roughly 7·8 D.
(*b*) Depth of bars	1·4 D + ·75 inch.
(*c*) Thickness of bars	·5 D + ·15 ,,
(*d*) Width between bars and thickness of valve-rod eye when gun-metal or bronze block	1·2 D + ·25 ,,
(*e*) Width between bars when cast-steel block and gun-metal liners	1·65 D + ·25 ,,
(*f*) Dia. of valve-rod eye (G.M. or bronze block)	1·55 × depth of link-bar.
(*g*) ,, ,, (steel block and G.M. liners)	1 × ,, ,,
(*h*) Length of sliding block	3 D to 3·75 D.
(*j*) Diameter of eccentric-rod pins . .	·85 D + ·5 inch.
(*k*) Length ,, ,, . . .	·7 D + ·4 ,,
(*l*) Diameter of drag-rod pins . . .	·7 × diameter of eccentric-rod pin.
(*m*) Length ,, ,, ...	1 × length of eccentric-rod pin.
(*n*) Diameter of bolt through ends of link bars	·35 × depth of link bar.

For proportions given by above rules, *see* Table LXV.

The links and all the pins and working surfaces should be thoroughly case-hardened before completion if not made of hard steel.

Position of suspension pins.—It is convenient to place the suspension, or drag-rod pins at the end, in the case of the slot link, but, when the engine has to work equally ahead or astern, they are better placed at mid-length of the link,—so as to coincide in position with the block-pin when the link is in mid-gear,—so as to give the least "slotting" motion.

The reversing lever when in mid-position was commonly parallel to the valve-rod, but it is better to incline it slightly away from the valve-rod, and so reduce the "slotting" when in ahead gear, whilst increasing it a little when in astern gear. The amount of inclination, measured on the arc, may be about one-eighth the length of the lever.

The drag-rods should, of course, be as long as they can conveniently be made, and care should be taken that the angle they make with the reversing lever is not too obtuse when at its maximum.

The position of the pin of reversing lever for the least amount of slotting can be found by placing the link in "ahead" gear and

Table LXV.—Proportions of Double-Bar Links.

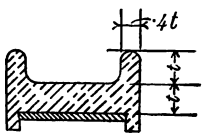
Diameter of valve-rod (in valve).	Centres of eccentric rod pins, (about)	Depth of bars.	Thickness of bars.	Width between bars, G.M. block.	Width between bars, steel block.	Diameter of valve-rod eye, G.M. block.	Diameter of valve-rod eye, steel block.	Length of block on bars, (3·5 D)	Diameter of eccentric rod pins.	Length of eccentric rod pins.	Diameter of drag-rod pins.	Length of drag-rod pins.	Diameter of end bolt.
	(*a*)	(*b*)	(*c*)	(*d*)	(*e*)	(*f*)	(*g*)	(*h*)	(*j*)	(*k*)	(*l*)	(*m*)	(*n*)
inches.	inches.	inches.	inches.	inches.	inches.	inches.	inches.	inches.	inches.	inches.	inches.	inches.	inches.
1½	12	2⅞	⅞	2	...	4⅛	...	5¼	1¾	1½	1	1½	1
1¾	14	3¼	1	2⅜	...	5	...	6	2	1⅝	1⅛	1⅝	1⅛
2	16	3½	1⅛	2⅝	...	5½	...	7	2¼	1¾	1¼	1¾	1¼
2¼	18	4	1¼	3	...	6¼	...	7¾	2⅜	2	1⅜	2	1⅜
2½	20	4¼	1⅜	3¼	...	6¾	...	8¾	2⅝	2⅛	1½	2⅛	1½
2¾	21	4⅝	1½	3½	4¾	7¼	4⅝	9½	2¾	2⅜	1⅝	2⅜	1⅝
3	23	5	1⅝	3⅞	5¼	7¾	5	10½	3	2½	1¾	2½	1¾
3¼	25	5¼	1¾	...	5⅝	...	5¼	11¼	3¼	2⅝	1⅞	2⅝	1⅞
3½	27	5⅝	1⅞	...	6	...	5⅝	12¼	3½	2⅞	2	2⅞	2
3¾	29	6	2	...	6½	...	6	13	3¾	3	2⅛	3	2⅛
4	31	6⅜	2⅛	...	6⅞	...	6⅜	14	4	3¼	2¼	3¼	2¼
4¼	33	6¾	2¼	...	7¼	...	6¾	14¾	4⅛	3⅜	2⅜	3⅜	2⅜
4½	35	7	2⅜	...	7⅝	...	7	15¾	4¼	3½	2½	3½	2⅜
4¾	37	7⅜	2½	...	8	...	7⅜	16½	4½	3¾	2⅝	3¾	2½

Notes.—In Naval work, owing to the higher speeds of revolution used, the proportions and surfaces are usually somewhat greater than in Mercantile work; the above Table gives good average proportions. For paddle engines the surfaces may be same as for screw engines, since little is saved by reducing them.

tracing the *locus* of the centre of the suspension pin when there is no slotting motion whatever. With the length of drag-rod as radius, draw an arc through the figure so traced so as to divide it thereby as uniformly as possible. The centre of this arc is the position of the lever pin. Carry out the same operation for the link in "astern" gear and the other extreme position of the lever end is found. From these two positions the centre of weigh-shaft can be located.

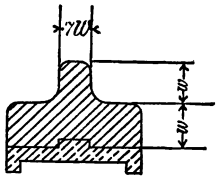
ECCENTRICS.

Eccentric Sheaves.—These should be made of hard and tough cast-iron, and, when made in two pieces, the small piece should be of specially strong iron or steel. The line of division should always be at right angles to the line through centres of shaft and of eccentric, and the key-way should be placed in the centre of the large piece. The



Bronze Strap lined with White-metal.

FIG. 36.



Cast-steel Strap with Bronze liner.

FIG. 37.

joint separating the two portions should also have a small step in it, on each side of the shaft, to prevent their shifting.

The bolts holding the two portions together may either have suitable nuts fitted into recesses in the large portion, or cotters placed where the sheave is perforated.

Eccentric straps.—These may be of good bronze for sheaves on shafts up to 10 inches in diameter, but for larger eccentrics they should be of cast-steel, with white-metal or bronze liners. In the mercantile marine good cast-iron, with or without white-metal linings, works quite satisfactorily.

The straps should always have side lips, as shown in Figs. 36 and 37, and when bronze liners are used they should have external projecting rings fitting into grooves in the cast-steel straps, as shown in Fig. 37.

When bronze straps are used, each half should be made thicker at the middle, to allow for wear; and whether bronze or steel is used, the strap should be stiffened by ribs as indicated in Figs. 36, 37, and 38,—one central rib being used for steel, and two side ones for gun-metal.

The general proportions of sheave and strap should be as follows :— the unit D being still the diameter of valve-rod, as found by Rule 166 :—

Rule 172, a to m.

(*a*) External diameter of sheave		1·2 × diameter of (shaft + throw of eccentric), or roughly 7·8 D.
(*b*) Breadth of sheave at strap	Mercantile	D + ·6 in.
	Naval	D + 1·25 ,,
(*c*) Thickness of metal round shaft	pierced sheave	·6 D + ·5 ,,
	lightened ,,	·45D + ·25 ,,
(*d*) ,, ,, at circumference	pierced ,,	·4 D + ·5 ,,
	lightened ,,	·4D + ·25 ,,
(*e*) Breadth of key		·55D + ·4 ,,
(*f*) Thickness ,,		·27D + ·3 ,,
(*g*) Diameter of bolt connecting portions of sheave		·45D + ·25 ,,
(*h*) Thickness of bronze straps at middle (ex. lips) (*t*)		·4D + ·5 ,,
(*j*) ,, ,, ,, sides ,,		·33D + ·5 ,,
(*k*) ,, cast-steel straps (*w*)		·3D + ·5 ,,
(*l*) ,, bronze liners for cast-steel straps.	$\frac{\text{Dia. of sheave}}{70}$	+ ·125 ,,
(*m*) Diameter of bolts connecting parts of strap		·5D + ·2 ,,

For proportions given by above rules, *see* Table LXVI.

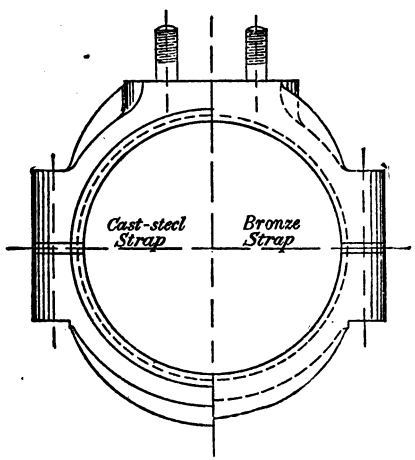


FIG. 38.

Table LXVI.—Proportions of Eccentric Sheaves and Straps.

Diam. of valve-rod (in valve).	External diam. of sheave (about)	Breadth of sheave at strap. (b)		Thickness of metal round shaft. (c)		Thickness of metal at circumference. (d)		Dimensions of Key. (e) (f)		Diam. of bolt connecting sheave.	Thickness of bronze straps. (h) (j)		Thickness of cast-steel straps.	Thickness of liners for steel straps.	Diam. of bolt connecting strap.
		Mercantile.	Naval.	Pierced	Lightened.	Pierced	Lightened.	Br'dth	Thickness.	(g)	Middle.	Sides.	(k)	(l)	(m)
ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.	in.	ins.	ins.	ins.	in.	in.	ins.
1½	12	2⅛	2¾	1½	1	1⅛	⅞	1¼	⅝	1	1⅛	1	1	5/16	1
1¾	14	2⅜	3	1⅝	1⅛	1¼	1	1⅜	13/16	1⅛	1¼	1⅛	1¼/16	5/16	1⅛
2	16	2⅝	3¼	1¾	1⅛	1⅜	1⅛	1½	⅞	1⅛	1⅜	1⅛/16	1⅛	⅜	1¼
2¼	18	2⅞	3½	1⅞	1¼	1⅜	1⅛	1⅝	15/16	1¼	1⅜	1¼	1⅝/16	⅜	1⅜
2½	20	3⅛	3¾	2	1⅜	1½	1¼	1¾	1	1⅜	1⅝	1⅜	1¾	7/16	1½
2¾	21	3⅜	4	2⅛	1½	1⅝	1⅜	1⅞	1 1/16	1½	1⅝	1⅞/16	1⅝/16	7/16	1⅝
3	23	3⅝	4¼	2⅜	1⅝	1¾	1½	2	1⅛	1⅝	1¾	1½	1⅞/16	7/16	1¾
3¼	25	3⅞	4½	2½	1¾	1⅞	1⅝	2¼	1 3/16	1¾	1⅞	1⅝	1½	½	1⅞
3½	27	4⅛	4¾	2⅝	1⅞	2	1¾	2⅜	1¼	1⅞	2	1 11/16	1⅝/16	½	2
3¾	29	4⅜	5	2¾	1⅞	2	1¾	2½	1⅜	1⅞	2	1¾	1⅝	9/16	2⅛
4	31	4⅝	5¼	3	2	2⅛	1⅞	2⅝	1 7/16	2	2⅛	1⅞	1¾	9/16	2¼
4¼	33	4⅞	5½	3⅛	2⅛	2¼	2	2¾	1½	2⅛	2¼	1 15/16	1 13/16	⅝	2⅜
4½	35	5⅛	5¾	3¼	2¼	2⅜	2⅛	2⅞	1½	2¼	2⅜	2	1⅞	⅝	2½
4¾	37	5⅜	6	3⅜	2⅜	2⅜	2⅛	3	1 9/16	2⅜	2⅜	2⅛	1 15/16	⅝	2⅝

As a check on the diameter of bolts in strap, it should be seen that the loads on them are not in excess of those given in Table XLIII.

Eccentric Rods.—These should be sound forgings of mild steel,—the double-eye at the link end being cut out of the solid forging. The diameter at the link end should be that given by,—

Rule 173.

Diameter of eccentric rod at link end = ·8 D + ·2 inch.

The diameter at mid-length of rod may be calculated in the same manner as a connecting-rod (Rule 81), but for direct-acting engines of ordinary construction,—(where the length of connecting-rod is not more than 2 × stroke for Naval engines, and 2¼ × stroke for Mercantile engines),—the rods may be made with a straight taper from end to end, and may have the following diameters at the ends next eccentrics:—

Rule 173a.

Diameter of eccentric rod at eccentric end . . . . . { **Naval**.....D
Mercantile...D + ·4 inch,

where D has the same value as before.

The studs by which the T end of the eccentric rod is secured to the strap should be of the same strength as the bolts for valve-rod caps (Rule 169), and for connecting the two parts of the strap (Rule 172*m*), and their diameter may be calculated in the same manner.

The cap-bolts at link end of rod should also, collectively, have the same strength as the valve-rod cap-bolts; but, as they will be considerably smaller, they must not carry so great a load per square inch of section at bottom of thread, and their diameter will be easily fixed by reference to Table XLIII.

JOY'S VALVE GEAR.

The general arrangement of one form of this gear is shown in Fig. 39. The method of laying down the centre lines is as follows:—

Take a point d (Fig. 40, page 197) on or near the axis of the connecting-rod, such that its extreme transverse vibration is about twice the full stroke of the valve (better rather more than less); and through its extreme positions d_1 d_2 draw zz. Then mark off e and e_1,—the extreme longitudinal positions of the point d,—and from these points draw two lines to meet in a point f on zz, the position of which is such that the angle efe_1 is not more than 90° (if there is room a less angle is better).

The point f should be controlled so as to move as nearly as possible on the line zz; this may be effected either by an "anchor" link 2,—which may be placed on either side of zz,—or by a sliding guide.

Next, on the valve-rod centre 5, mark off g_1 and g_2, so that gg_1 is

equal to lap + lead for the top or back end of cylinder, and gg_2 equal to same for crank end of cylinder. Then take a point j on the link $e_1 f$ so that $e_1 j$ is about $3 \times dd_1$, and draw jg_1, cutting zz in m; m will be the fulcrum of the lever 3, and will also be the centre of the curved reversing block in which the fulcrum slides. When swing links are used (as in Fig. 39), in lieu of a curved block, the arcs

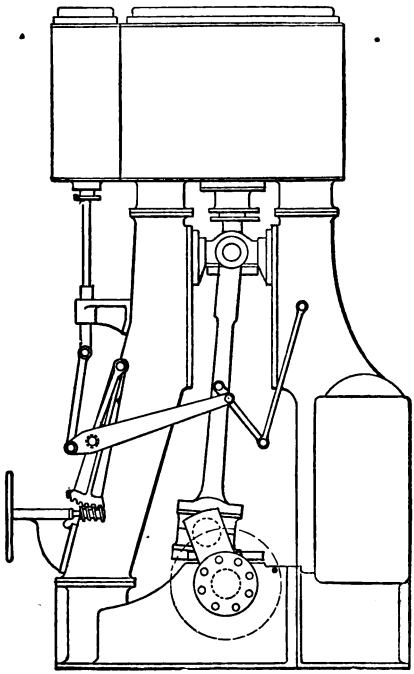


FIG. 39.

described by the link ends, when in ahead and astern positions, must intersect in M.

The position assumed for j in the first instance is approximate only, and may not be quite correct; its position must be such that the point m in the lever jmg_1 moves to an equal distance on either side of the centre of the reversing block,—also marked m. If not at first quite correct, it will generally be obtained on a second trial.

When the gear is correctly set out, the act of moving it from ahead to astern gear, or to any intermediate position, should leave the lead unaltered.

The valve-rod link, which is attached at g, may have any convenient

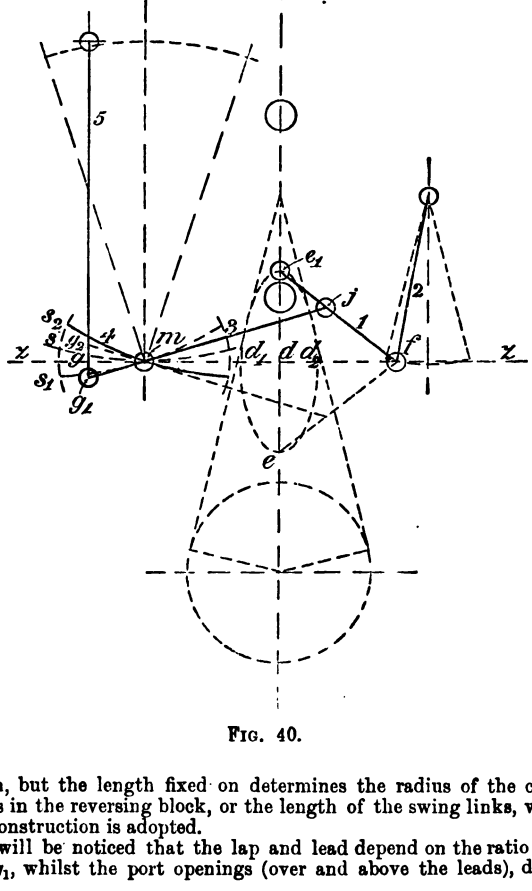


FIG. 40.

length, but the length fixed on determines the radius of the curved guides in the reversing block, or the length of the swing links, whichever construction is adopted.

It will be noticed that the lap and lead depend on the ratio of jm to mg_1, whilst the port openings (over and above the leads), depend

only on the angle to which the reversing block is canted, and will be about $^4/_5$ths of the distances ss_1 and ss_2 respectively.

Considerable deviations from the above-described positions and proportions may be made without materially affecting the nature of the results.

Where the distance from centre of piston-rod to centre of valve-rod is different for the different cylinders (H.P., M.P., L.P.), the valve-rod link may be inclined as much as 1 in 12, but beyond that the point m should be moved, and the lengths jn and mg_1 altered.

For complete investigation of Bremme's or Marshall's gear see Mr Seaton's "Manual of Marine Engineering."

REVERSING GEAR.

For all ordinary types of engine, when size renders hand-gear too slow in its action, the simplest and most efficient form of reversing engine is the direct-acting steam cylinder, controlled by a cataract brake. A very perfect form of this gear has long been associated with the name of Messrs Brown of Edinburgh; by the adoption of valve-gear of the "hunting" type they have produced a machine which is handled with the greatest ease and certainty, and which is, at the same time, quite automatic in its movements. The action of the gear is as follows:—On moving the hand-lever through any fraction of its travel, the steam valve is displaced, or opened, a corresponding amount causing the piston to move; likewise as the gear begins to move, the steam valve begins to return to its central or closed position; consequently when the piston has travelled through the fraction of its stroke corresponding with the original movement of the hand-lever, the valve is again central, and the gear at rest.

Direct steam gears should never be fitted without the brake cylinder, as, in careless hands, the steam-gear is apt to "take charge," overpower the controlling hand-gear, and "carry away" some of the brackets, &c.

Another very excellent prime mover, which is adaptable to almost any type of engine, is that known as the "all-round" gear. In this arrangement a connecting-rod from the main lever on the weigh-shaft takes hold of a crank-pin on the side of a worm-wheel, whose worm is revolved by a small reciprocating steam engine.

The great advantage of this gear is that it has no "stops," but can travel round continuously, without risk of carrying anything away, and is therefore perfectly safe in even the most careless hands. It has also the further advantage,—resulting from the crank and connecting-rod principle embodied,—of exerting its greatest power at the extreme positions of links, so that it readily starts to reverse when required and is easily controlled when notching-up is desired. The small engine is sometimes made with two cylinders and cranks to be automatic, and is also sometimes made reversible; but these are both very doubtful improvements.

In all steam reversing gears, arrangements should be made to prevent the accumulation of water in the cylinders of the small engine, in order that they may be in condition to start instantly when the hand-lever is moved.

Reversing gear weigh-shafts.—These shafts must be not only strong enough for their work, but also fairly rigid under it, and capable of standing any bending stresses that may come upon them.

The following formula gives diameters in accordance with the best modern practice:—

Rule 174.

$$D = \sqrt[4]{\frac{d^2 \times \text{No. of gears} \times L \times l}{8}} + \cdot 25,$$

where D = diameter of weigh-shaft in inches;
d = effective diameter of valve-rod in inches;
l = length of drag-rod lever in inches;
L = length of weigh-shaft in feet.

For triple engines of ordinary proportions the above formula gives a diameter of weigh-shaft about equal to twice the effective diameter of the valve-rods,—slightly larger if both forward and after valve-boxes are "outside," and slightly smaller if both are "inside."

Another, a simple and reliable rule for weigh-shafts, is as follows:—

Rule 174a. $\quad \text{Diameter} = \sqrt{\frac{\text{N.H.P.} \times \text{Revs.}}{F}}$

N.H.P. is that given by Rule 7.

F	for two-crank	engines,	1000.
F	,, three-	,,	900.
F	,, four-	,,	800.

The designer must, of course, use his discretion and make suitable modifications where the weigh-shaft bearings cannot be placed close to the various levers, or the steam gear lever close to the middle valve-gear in a three-crank engine, or between the two middle gears in a four-crank one.

In some cases the weigh-shaft may be made tapering to the ends, but, as a rule, it is better to use a practically parallel shaft, and, if weight is of great importance, make it hollow, as is commonly the case of large Naval engines; the necessary stiffness is then maintained without difficulty, and there is no fear of the bending.

Large weigh-shafts should be made in two pieces connected by solid flange couplings, and the main lever may be placed between the coupling flanges and secured by passing the coupling bolts through it. "Enlargements" should be provided on the shafts for such other levers as must be fixed at any distance from the ends.

Size of reversing engine cylinders. — For ordinary triple engines as described above, this may be obtained from the general relation,—

The steam cylinder of the direct "push and pull" gears should be sufficiently large to act freely with steam at a pressure considerably lower than "boiler pressure."

Rule 175.—Capacity of cylinder should be the capacity of L.P. cylinder ÷ 96.

For all-round gearing with the usual proportion of worm and wheel and length of levers.

Rule 176.—Capacity of a single cylinder should be practically that of the L.P. cylinder ÷ 600.

In their fear of giving too little power in steam reversing gears, there is little doubt that engineers often run to the other extreme, and give much more than necessary, even when two-thirds boiler pressure is assumed. The values assumed for the constants in the above formulæ give ample size.

With "all-round" gears the small engine should not make less than 15 revolutions in moving the gear over, and where possible 20 should be allowed, and up to 25 in large engines.

The pitch of the worm-wheel teeth should never be less than 1½ inches, and for large engines 2½ inches is usual; they should be machine-cut, and should have no more clearance than is absolutely necessary.

The worm is usually forged solid with its spindle, and may have a diameter at pitch given by,—

Rule 177.

Diameter of worm at pitch circle may be . 3 × pitch of teeth.

Rule 177a.

Width of face of worm-wheel ,, . 2× ,,

Rule 177b.

Length of worm (actual thread) ,, . 3× ,,

STEAM TURNING GEARS.

The turning gear now universal is the double worm arrangement,—in which the second, or auxiliary, worm shaft is coupled direct to the crank-shaft of the small turning engine, which makes from 1200 to 2000 revolutions for one turn of the main engines, according to size. The size of cylinder, or cylinders, of turning engine should be such that, with say 50 lbs. of steam, the main engine can be turned one revolution in from 5 to 8 minutes, according to size, which gives,—with the above-mentioned ratios of gearing,—a speed of 250 revolutions per minute for the turning engine.

In the Navy the turning engines have usually two cranks, and are made reversible by means of links or a reversing valve, but one is quite sufficient, and a loose eccentric is all that is required for reversing.

In small vessels, where a special turning engine is not considered

necessary, the auxiliary worm-shaft may be driven from a donkey pump or other auxiliary engine, by means of a pitch chain and wheels.

For ordinary three-crank triple and quadruple engines, the sizes of cylinder for turning engine should be such as will satisfy the following relation,—the boiler pressure being assumed as 50 lbs. in all cases:—

Rule 178. $$\frac{A \times T}{D^3} = K,$$

where A = area of piston or pistons of turning engine, in sq. inches;
T = travel of piston or pistons, in feet, while turning main engines through one revolution;
D = diameter of crank-shaft of main engines:
K = 14.

Rule 179.—In general practice the capacity of the cylinder of a turning engine is = the capacity of the L.P. cylinder of the main engine ÷ 700.

In Naval work the main wheels are commonly of cast-steel with gun-metal rims, but a plain cast-iron wheel, gearing with a hard bronze worm, gives quite as good results and is much cheaper to fit.

However carefully a worm-wheel may be moulded there is sure to be more or less distortion in cooling, and it is therefore the cheapest in the long run to machine-cut the teeth.

The teeth should not be made too fine a fit; there is no objection to a good clearance.

The usual pitch for the teeth of the main wheel is as follows:—

For 6-inch to 8-inch crankshaft . .	2-inch pitch.
8½ ,, 11 ,, ,, . .	2¼ ,,
11½ ,, 14 ,, ,, . .	2½ ,,
14½ ,, 18 ,, ,, . .	3 ,,
above 18 ,, ,, . .	3½ ,,

When calculating the strength of the teeth, the number in gear at one time should not be taken at more than two.

Rule 180.—The length of tooth should not exceed ·6 × pitch, viz. ·26 × pitch outside the pitch circle, and ·34 × pitch within.

It is scarcely necessary to say that the pitch should be an even quarter or half inch, and that the diameter of the wheel must then be made such as will give the required number of teeth; also when the wheel is made in halves the number of teeth should be even.

The pitch of the auxiliary worm-wheel is usually 1½ inches to 2¼ inches.

Rule 181.—The width of face of the wheels may be 1·75 × pitch of teeth when a solid worm is used, and 2·1 × pitch when the worm is loose on the spindle.

The diameter of the pitch cylinder of the worm may be,—

Rule 182.

When worm is forged with spindle . . 2·4 × pitch of teeth.

Rule 182a.

When worm is loose on spindle { forged steel . 3 × ,, / bronze . 3·3 × ,,

Length of worm (actual thread) may be 3 × pitch of teeth.

Wheels up to 3 feet diameter are best made with a continuous plate, or disc, in lieu of arms; larger wheels may have six radial ribs on each side of the disc (the flanges counting as two, when the wheel is in halves), and the disc may then be lightened by cutting six circular, or nearly circular, holes in it.

The thickness of rim, of disc plate, and of ribs, may be same as thickness of tooth at pitch line.

Where possible the main wheel should be placed on a coupling. When cast-steel brackets or bearings are used for the worm-shafts they should be bushed with bronze or white metal.

SCREW PROPELLERS.

Ships may have any number of screw propellers, and in the past some have had as many as six. In practice the following holds good:—

(i) **Single screw** at the stern for all small ships of sufficient draught of water—all cargo steamers of moderate speed.

(ii) **Two screws**, generally side by side, and called *twin*, for all fast steamers, Naval ships, and for cargo steamers of light draught or high speed.

Two screws, one at the bow and one at the stern, have been tried with success in tugs and other high-powered, small, short craft which are required to maintain full power in a sea-way, inasmuch as when pitching one screw is always fairly immersed.

(iii) **Three screws** have been employed with reciprocating engines in ships of large power and moderate draught—chiefly in Naval ships of foreign powers. Since the introduction of the turbine three screws have been common in ships of all sizes as convenient to the turbine installations and conditions of high revolution with direct drive.

(iv) **Four screws** were originally tried on light-draught ferry-steamers required to move with either end first. In the Navy and Atlantic Service ships of very large size and high speed propelled by turbines have four screws. In the latter case the screws are in pairs, the inner pair being just forward of the rudder, as in twin-screw ships; the outer pair are one on each side, well forward and clear of the feed to the others, and near enough to the skin of the ship to permit of a short outer fin to carry the screw shaft.

The single screw is of course the least expensive, but it is not so safe as the twin screws, one of which will propel the ship when the other is out of action, and with both in action she can be steered in case of disablement of the rudder or its gear.

It is better to have a small screw well immersed than a large one

with its tips near the sea surface, consequently when the draught of water of a ship is limited and not much more than the diameter of the single screw, two smaller screws will give a higher propulsive efficiency; and in the case of very high power three well-immersed screws will be better than two larger ones insufficiently submerged. When the draught of water is so light that no screw of reasonable diameter can be fitted submerged, then for good efficiency the screw race must be covered in so that when under way the screw submerges itself. This has been done most successfully by Sir J. Thornycroft and Mr Yarrow, the latter using a hinged flap which can be lowered and come into action when the ship is light aft; and when loaded deep enough not to require it the flap is raised out of water.

Screw propellers may have any number of blades; from one to six have been experimented with in actual practice. One blade is theoretically highly efficient; in practice it is objectionable for obvious reasons, chiefly for being out of balance, and its centre of effort revolves and causes complications.

Two blades balance, but their centre of effort has an orbit due to the difference of immersion of the blades during revolution, with the usual consequence of considerable vibration. With deep immersion the evil is not so great, and in smooth water the two-bladed screw has a high efficiency. Such screws are now only fitted to small craft, to boats driven by irreversible engines, and to sea-going ships which have to do a considerable amount of sailing. In the latter case the screw is either raised out of water or its blades moved into a fore and aft position, so as to offer little "head" resistance, by such mechanical means as employed to alter the pitch or reverse the blades' angles.

Three blades is the commonest for twin- and multiple-screw ships, and when well immersed is the most efficient for all sea-going ships. With the small diameter permissible with direct-driven turbines, there was difficulty in designing a screw blade with sufficient surface with only three, consequently four had to be resorted to for large-powered ships. With geared turbines there is no difficulty.

Four blades is the common practice for all single-screw ships, for large merchant ships of full power engaged in Atlantic service, and for twin-screw warships of large power and comparatively high draught, as well as for those other warships and express steamers with three and four screws of small diameter and large surface.

The shape of the screw blade is now generally an approximation to an ellipse, but in the mercantile marine there is still a tendency to what was called pear-shape—really an ellipse with truncated end—that is, square tipped. With the small diameter large surface screw it is much broader at the tip and shaped like a Japanese fan; and often the blade is circular, or circular with the tips cut square. Experience has led engineers into the fashion of having the maximum breadth of blade in the middle of small diameter screws, and at about two-thirds of the length from the tip with ordinary ones. In a general way the maximum breadth may be at a distance equal to a quarter of the diameter from the axis.

The section of the blade was generally such that the acting face is a true helical surface and the back curved so that any section flattened would be a segment of a circle. Experiments on model screws of a most exhaustive character by Professor D. W. Taylor and Capt. Dyson, U.S. Navy, have proved that while these are good sections they are by no means the best; that on the whole a more ship-shape (horizontal or water-line) section is the best, so that the maximum thickness is much nearer the leading than the following edge, and the entering angle more obtuse than was formerly thought good.

The material of propellers is a much more important matter now than formerly, as the high rate of revolution obtaining, especially with turbines, sets up centrifugal forces of great magnitude, and the erosion due to the action of sea-water on the surface is severe. Hence it is now necessary in express steamers to have the screw blades made of material which has a very high tensile resistance and a surface which will not easily corrode and permit of erosion.

Cast-iron is still used largely in the mercantile marine, and for cargo ships and those working at a moderate rate of revolution is quite satisfactory inasmuch as it is cheap, is strong enough, and will, bar accidents, last six years with common care. If it receives a heavy blow a portion of the blade is broken clean off, leaving the screw to revolve freely, so that when back in port the propeller may be removed and have a new piece "burnt" on; or, in case of a loose-bladed screw, a new blade can be shipped.

Bronze of kinds, mostly high tensile zinc and copper alloys with some "doctor" metal added, are in general use for express steamers, Naval ships, and other craft where cost is not so important a factor as high efficiency. The bronze screw is always thinner than cast-iron, has a smoother surface, and is cast truer to pitch, and wastes very little under the action of sea-water; moreover, it is much lighter to handle and carry. Phosphor bronze has also been used for blades, and from its high elastic limit and smoother surface has advantages, but it is not so tough.

Naval bronze or gun-metal is now seldom, if at all, used, as it is more costly and has a much lower elastic limit.

Steel castings were at one time used for large ocean-going express steamers; blades of this material were generally cast separate and had a very rough surface, which was not improved by the action of sea-water; they were seldom true to pitch and often very badly out from tip to root. Corrosion was rapid, so that their working life under favourable circumstances was only four years, and all attempts to prevent destruction proved failures.

The dimensions of a screw propeller depend on the circumstances of each case. Sometimes they have to conform to the requirements of the engines, and sometimes the engines must be designed to suit the screws. Oftener still they are the result of compromise between the demands of both.

The diameter of a screw is governed by the draught of water of the ship and the form of the stern lines, the exceptions to the latter have

been already cited. The screw should have its top blade immersed to the extent of 15 per cent. of its diameter to be efficient on ocean service, and 10 per cent. for smooth water work. A ship having fine water-lines in the "run" will permit of a better "feed" to the screw and have wake currents less extensive than one with a full "run," consequently she may have a screw of smaller diameter. In the same way a twin-screw ship may have smaller screws *pari passu* than a single-screw ship of the same form. All other things being equal, the power delivered to the screw per revolution, that is $a\frac{\text{I.H.P.}}{\text{R}}$ is the criterion of dimensions. The screw has with that power to project a column of water, so that by its reaction an adequate thrust on the ship is produced; within reasonable limits it may be taken that if D be the diameter of the column, and L its length, then

$$\frac{\pi D^2}{4} \times L = \text{constant}; \text{ that is } D^2 \times L \text{ is constant.}$$

In other words, that D^2 varies inversely as L, so that if the screw's diameter increases the pitch may be less and *vice versa*. Consequently for a given power there may be a great variety of dimensions for the screw. There may be one with small diameter and big pitch, or one of large diameter and smaller pitch.

Pitch ratio is the expression for the relation of pitch to diameter, and may be anything the designer chooses, but practice has shown that $\frac{P}{D}$ should not be much less than 1·0, and propulsive results are better when it is greater than 1, and the best are attained with ordinary ships when the pitch ratio is from 1·2 to 1·6.

The slip of a screw is of course a real thing as well as an apparent one. That is, the speed of the ship through the water is less than the multiple of the revolutions and acting pitch, and the difference between them is the *apparent slip*. *The real slip* is the movement of the water imparted to it by the propeller through the surrounding water, and what that is cannot be measured; in any and every case it is more than the apparent, and with a bluff ship with enormous wake currents it is much more, and of course is always positive. *Negative apparent slip* is often observed with such ships when they have screws of large diameter and surface.

Surface of Screw (acting) is that of the blades from the boss to the tip acting on the water as the screw turns; and its area is that actually measured on the blade.

Projected surface is the area projected on a plane transverse to the screw's axis from the acting surface, and is often taken by preference as the criterion of the screw capability.

Thrust.—If V is the velocity of the stream of water issuing from the ship, whose own velocity is v in feet per second; then velocity of the stream with respect to *still water* past which the ship is moving will be $(V - v)$. This is the *real slip*. Then:—

Rule 183. Real slip per cent. of velocity of flow $= \frac{(V-v)}{V} 100.$

Rule 184. The mass of sea-water will be

$$\frac{\pi D^2}{4} \times V \times \frac{64}{32} = 1{\cdot}571\ D^2 \times V.$$

The momentum of the stream $= 1{\cdot}571\ D^2V(V-v) =$ thrust in lbs.

The work done $= 1{\cdot}571\ D^2V(V-v)v$ in foot-lbs.

Rule 185. Horse-Power $= \frac{1{\cdot}571\ D^2V(V-v)v \times 60}{33{,}000} = \frac{D^2V(V-v)v}{350}.$

The diameter of the screw may be determined by this formula as follows:—P is the pitch, R the revolutions per minute; $\frac{V-v}{v} = f$, a fraction of V; E is the efficiency of the screw and engine, that is the ratio of actual work of propulsion to the I.H.P. developed by the engine; or in the case of turbines is the ratio to the S.H.P., and therefore the efficiency of the screw only. Then the following holds good,

$$\text{I.H.P.} = \frac{D^2 \times (P \times R)^3 (f - f^2)}{350\ E}.$$

Rule 186. Diameter of Screw in feet $= \sqrt{\frac{\text{I.H.P} \times 350\ E}{(P \times R)^3 \times (f - f^2)}}.$

If the real slip be taken as $x = 0{\cdot}2P_c + f$ for twin screws and $0{\cdot}18\ P_c + f$ for single and centre screws of three-screw ships, where f is as before and P_c is the prismatic co-efficient of the ship's submerged hull, then

Rule 186a. Diameter of Screw in feet $= \sqrt{\frac{\text{I.H.P.}}{x(1-x)} \times \left(\frac{450}{P \times R}\right)^3}.$

A good practical method of determining the diameter of a screw suitable for a particular ship is by means of

Rule 186b. Diameter of Screw in feet $= z \times P_c \sqrt[3]{\frac{\text{I.H.P.}}{R}}$

For single screws ordinary,	$z = 7{\cdot}25$.	Ocean-going express,	$7{\cdot}61$.
,, twin ,,	$z = 6{\cdot}55$.	,, ,,	$6{\cdot}88$.
,, quadruple ,,	$z = 6{\cdot}25$.	,, ,,	$6{\cdot}51$.
,, turbine driven centre,	$z = 6{\cdot}55$.	,, ,,	$6{\cdot}88$.
,, ,, wing,	$z = 5{\cdot}75$.	,, ,,	$6{\cdot}04$.

A rough and ready method suitable to cargo steamers of good draught.

Rule 186c. Diameter of Screw in feet $= z\sqrt{D \times S}$

D is the diameter and S the stroke of L. P. piston in feet.
$z = (2{\cdot}4 + P_c)$ for twin screws and $(2{\cdot}7 + P_c)$ for single screws.

Pitch of screw is now generally uniform throughout, so that the propeller is said to be a *true screw*. If the screw is of small diameter for the power delivered to it, there is of necessity a larger slip ratio than one with larger diameter. Slip ratio is a variable factor, and may be chosen by the designer to suit the other conditions. It is now known, as the result of the very many and carefully made experiments, what the efficiency is of any particular screw at different slip factors; or with the same slip factor the efficiency with different pitch ratios. Curves of efficiency, with these variables, may be set out and a choice made to suit the revolutions of the engines. In the mercantile marine, with reciprocating engines an *apparent slip* of 10 per cent., and with turbines 15 per cent. is considered satisfactory. Bluff cargo boats seldom show so high a ratio, 5 per cent. being usual, and negative slip not unusual with large screws.

If S is the speed of the ship in knots per hour, R the revolutions per minute, and s the slip in knots,

$$\text{Pitch} = \frac{(S+s)\times 6080}{60\ R}.$$

If x is the slip per cent. of speed of screw, that is of $\frac{P\times R\times 60}{6080}$,

then $$S = \text{speed of screw}\left(1-\frac{x}{100}\right).$$

Rule 187. **Then Pitch** $= \frac{S}{R}\times\frac{10{,}133}{100-x}$ feet.

By referring to Rule 186 the slip for any set of conditions can be estimated and the pitch of screw calculated accordingly, or the pitch may be assumed from engine conditions and the diameter calculated by that rule with the assumed apparent slip as f.

The surface of the screw is after all the most difficult to determine, that the very highest propulsive efficiency possible may be obtained with given diameter and pitch. The tendency of any increase in blade area is to add to the resistance to turning, as the decrease is to an augmentation of apparent slip without gain in propulsive efficiency—in other words, the actual column of water is not thereby increased with increase in revolutions. To go to extremes, a propeller of suitable diameter and pitch may fail to propel efficiently, inasmuch as the blade surface is insufficient to hold against the water, the blades being simply whirled through it. Starting with such a screw, any addition to blade surface tends to avoid this plunging through the water, so that the efficiency is improved; each addition makes a further improvement, but not at so great a rate; finally, the further increases produce no improvement, and the last addition detracts from efficiency. Breadth of blade, too, has considerable influence on efficiency of surface, so that with the minimum surface on the fewest blades there is the maximum breadth of blade and the greatest efficiency. The following is in accordance with good practice:—

Rule 188. Area of Acting surface in sq. ft. $= Q \sqrt{\frac{\text{I.H.P.}}{\text{Revs. p. m.}}}$.

$Q = P_c \times M$ where

For four-bladed screws, single, $M = 20{\cdot}0$. For twin screws, $15{\cdot}0$.
,, three ,, ,, $M = 19{\cdot}0$. ,, ,, $14{\cdot}3$.
,, two ,, ,, $M = 17{\cdot}5$. ,, ,, $13{\cdot}1$.

Maximum breadth of blade.

Rule 189. Maximum Breadth in inches $= K \sqrt{\frac{\text{I.H.P.}}{\text{Revs. per min.}}}$

where $K = 14$ in four-bladed screws; $K = 17$ in three-bladed; and $K = 22$ in two bladed. For turbines K is much higher when the diameter is so limited.

Thickness of blades in cases of very high rate of revolution and large power should be calculated most carefully by estimating the tension due to centrifugal force and the tension on the metal near the face due to the bending moment on that section from the reaction producing thrust.

In designing a screw it is not unusual to estimate a certain hypothetical thickness at the axis and get the longitudinal section by drawing a line from the tip to the points on the axis; then taking the thickness at the tip from practical considerations, draw a line parallel to the face line at a distance equal to this thickness till it meets the former line. This hypothetical thickness may be calculated by the following:—

Rule 190. Thickness of blade at axis $= \sqrt{\frac{d^3}{n \times b}} \times X.$

The value of X for cast-iron is $6{\cdot}3$; for gun-metal (Admiralty bronze) $3{\cdot}2$; for strong bronzes $2{\cdot}35$; and for steel $2{\cdot}5$.

d is the diameter of the tunnel shafting or that required to resist torsion, and may be taken as $\sqrt[3]{\frac{\text{I.H.P.}}{R}} \times H.$

The value of H for triple and quadruple compound engines may be taken as $5{\cdot}8\sqrt{p}$, and for turbines 65. Substituting

Rule 191. Thickness of blade at axis $= X \sqrt{\frac{\text{I.H.P.}}{R} \times \frac{H}{n \times b}}$,

where n is the number of blades and b the breadth at the root in inches near the boss, and p the boiler pressure.

But it may be preferred to estimate the thickness of blade at a point where it actually exists and has a definite breadth, say at $1\frac{1}{2}$ diameters of shaft from the axis. In this case the same rule applies, but the value of X is different.

Rule 191a, when thickness is estimated at $1{\cdot}5 \times d$ from the axis, then for cast-iron $X = 4$; 2 for Admiralty bronze; $1{\cdot}6$ for cast-steel; and $1{\cdot}5$ for forged steel and strong bronzes.

The thickness of blades at the tip should be as follows:—

Thickness of blade at tip—Cast-iron, . . ·04D + ·4 inch.
,, ,, Cast-steel, . . ·03D + ·4 ,,
,, ,, Gun-metal, . . ·03D + ·2 ,,
,, ,, High-class bronze, . ·02D + ·3 ,,
where D is diameter of propeller in feet.

The thickness of blade for loose-bladed Naval propellers may be taken roughly by :—

Rule 191b. $$t = \sqrt{\frac{T \times L}{d \times K}} + \cdot 25,$$

where t = thickness of blade in inches at surface of boss sphere ;

T = indicated thrust per blade = $\frac{\text{I.H.P.} \times 33{,}000}{\text{Pitch} \times \text{revs.} \times \text{No. of blades}}$;

L = ·6* × total length of blade (flange joint to tip) ;

d = diameter of blade flange ;

K = co-efficient = { 800 for Admiralty bronze ; 950 for strong bronze.

Table LXVII.—Thickness of Propeller Blades.

Dia. of shaft in inches.	Solid cast-iron, 4 blades.		Loose bladed propellers.					
	Breadth of blade = length of boss = 2·7 × dia. of shaft.	Thickness of blade (at centre of shaft.)	Breadth of blade = ·8 × dia. of flange = 1·8 × dia. of shaft.	Thickness of blade at centre of shaft.				
				4 blades cast-iron.	4 blades steel or bronze.	3 blades steel or bronze.	4 blades gun-metal.	3 blades gun-metal.
inches.	inches.	inches.	inches.	inches.	inches.	inches.	inches.	inches.
6	16	4¼	10¾	5	3	3½	3½	3⅞
7	19	4¾	12½	5¾	3½	3⅞	3⅞	4½
8	21½	5½	14½	6½	3⅞	4½	4½	5⅛
9	24	6	16¼	7¼	4⅜	5	5	5¾
10	27	6⅝	18	8	4⅞	5½	5½	6¾
11	30	7¼	19¾	8¾	5¼	6⅛	6⅛	7
12	32½	7⅞	21½	9½	5¾	6⅝	6⅝	7⅝
13	35	8½	23½	10¼	6¼	7⅛	7⅛	8⅛
14	38	9	25¼	11	6⅝	7⅝	7⅝	8¾
15	40½	9⅝	27	11¾	7⅛	8¼	8¼	9⅜
16	43	10¼	28¾	12½	7⅝	8¾	8¾	10
17	46	10⅞	30½	13¼	8	9¼	9¼	10⅝
18	...	...	32½	14	8½	9¾	9¾	11¼
19	...	...	34¼	...	8⅞	10¼	10¼	11¾
20	...	...	36	...	9⅜	10¾	10¾	12⅜

* For ordinary Griffiths form of blade ; if blades are broad at tip this quantity should be increased, and *vice versa*.

If cast-iron of lower than Admiralty quality (tensile strength 9 tons per square inch) be used, suitable increase in thickness of blade must be made.

If breadth of blade be made less than length of boss, thickness must of course be so increased that value of $b \times t^2$ shall always agree with that found from above Table.

A good method of setting out the longitudinal section of the blade is as follows :—

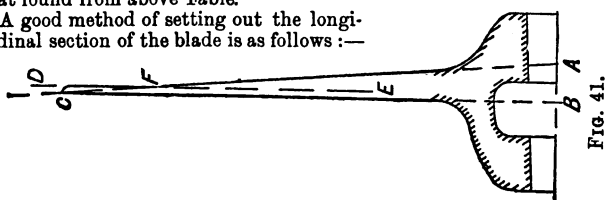


FIG. 41.

Determine AB (Fig. 41) from Table LXVII., or by Rule 191; draw AC so as to give no thickness at tip; set off thickness at tip (from Table LXVIII.), and draw DE parallel to CB (at right angles to shaft axis), and then fill in the back of the blade near F so as to make DF die gradually into AC.

Table LXVIII.—Thickness of Propeller Blades at Tip.

Diameter of Propeller in feet.	Thickness at tip, in inches.		
	Cast-iron ·04D+·4	Cast-steel. ·03D+·4	Bronze ·03D+·2
7	1 1/16	5/8	7/16
8	3/4	5/8	7/16
9	3/4	11/16	7/16
10	13/16	11/16	1/2
11	13/16	3/4	1/2
12	7/8	3/4	9/16
13	15/16	13/16	9/16
14	15/16	13/16	5/8
15	1	7/8	5/8
16	1 1/16	7/8	11/16
17	1 1/16	15/16	11/16
18	1 1/8	15/16	3/4
19	1 3/16	1	3/4
20	1 3/16	1	13/16
21	1 1/4	1 1/16	13/16
22	1 1/4	1 1/16	7/8
23	1 5/16	1 1/8	7/8
24	1 3/8	1 1/8	15/16

Studs or screws for attaching blades to boss.—These are usually of one of the stronger bronzes, for bronze propellers, and of mild steel for cast-iron or cast-steel propellers. They should be of size given by following formula :—

Rule 192. $$a \times N \times r = \frac{T \times L}{K}$$

where a = area of one stud or screw at bottom of thread (sq. inches);
N = number of studs or screws for one blade (usually 7 to 11);
r = radius of studs or screws (in inches);
T = indicated thrust per blade = $\frac{\text{I.H.P.} \times 33{,}000}{\text{Pitch} \times \text{revs.} \times \text{No. of blades}}$;
L = ·6 × total length of blade (flange joint to tip) in inches;
K = 1700 for studs, steel and strong bronzes; 1400 for Naval bronze bolts.

For ships constantly running in rough waters or in crossing the Atlantic, these factors should be 20 to 25 per cent. larger. With cargo steamers, which have often to steam when light at good power, and all of it taken by one blade, there is need of the same modification. On the other hand, ships in smooth water service, or only occasionally exerting full power, may have much lighter blades, so that K in that case may be 10 per cent. less without risk.

The weight of propellers may be estimated with a good approximation to the actual by the following :—

Rule 192a.

$$\text{Weight in cwt.} = \frac{\text{Surface in sq. ft.} \times \text{thickness of root in inches}}{Y}.$$

The value of Y for solid cast-iron screws is 4·5.
" " " bronze " 3·8.
" " " steel " 4·0.
The value of Y for loose-bladed cast-iron screws is 3·0.
" " " cast-steel " 2·8.
" " " bronze " 2·5.

Propeller Bosses when cast separately from the blades are often of the same material, but as the stresses when fitting on the shaft and by the action of the blades when at work are often severe, they are better of a somewhat tougher metal. Thus cast-iron bosses should be of a tough mixture, of which hematite or even steel is a component part; when weight is not of great moment such tough cast-iron bosses of ample thickness can be used with bronze and steel blades, and steel bosses, which can be made now quite cheaply, do very well with bronze blades, and are of course much cheaper than bronze ones.

The bosses of solid propellers are usually egg-shaped ; those for loose blades are spherical, and sometimes with a tapering tail to avoid cavity at their rear at high speed of ship.

Rule 193.

The diameter of boss for loose blades = $0{\cdot}9\sqrt{\text{diameter of screw.}}$

The length of boss for three and four blades is usually 0·75 its diameter.

The bosses of solid propellers are usually in diameter about ⅛ to ⅛ the diameter of the screw and 2·5 to 2·75 diameter of shaft in length.

The bolts or studs for securing blades are usually of strong bronze, but sometimes studs are of steel with bronze cap nuts.

The tension on them due to centrifugal force $= \frac{Wv^2}{32r}$.

W is the weight of a blade in lbs., v the velocity in ft. per sec. at centre of gravity of blade, whose distance from the axis is r feet.

If R be the revolutions per min. and the pitch P, then

$$v = \frac{R}{60}\sqrt{P^2 + (2\pi r)^2}.$$

The diameter of blade bolts whose number is n may be calculated from these rules and found as in practice by the following :—

Rule 194. Diameter of blade bolts $= \frac{d \times z}{n}$,

where $z = 1{\cdot}6$ for a three-bladed screw and 1·3 for a four-bladed, d being as before the equivalent diameter of shaft for torque.

Blade flanges.—These should be of the following proportions :—

Rule 195. Diameter of flange = 2·25 × diameter of tail shaft.

Rule 195a.

Thickness of flange $= \begin{cases} \cdot 85 \times \text{diam. of stud for G.M., bronze, or steel.} \\ 1{\cdot}4 \times \quad ,, \quad ,, \text{ cast-iron.} \end{cases}$

Proportions of propeller boss.—The bosses of solid cast-iron propellers may also be of the following proportions :—

Rule 196. Length of boss = 2·7 × diameter of tail shaft.

Rule 196a. Diameter of boss = 2·7 × ,, ,,

Rule 196b. The fore and aft section of the boss should be oval,—the principal radius being ·8 × diameter of boss.

The length of the tapered bore may be divided into three approximately equal parts, of which the two end ones will bear on the shaft, while the central one will be cored back.

The thickness of "shell" of the central, or cored out, portion of the boss should be,—

Rule 196c. Thickness of boss = ·65 × thickness of blade at shaft axis.

Where the blade flanges are all within the sphere of the boss,—as is usual in Naval propellers,—the extreme diameter of boss is about $0{\cdot}9\sqrt{\text{diameter of propeller}}$; the length of such a boss should be the same as that of a cast-iron boss (Rule 196).

As previously mentioned under "Tail shafts," the taper to which the boss is bored out should be about 1 inch on the diameter for each foot of length, but never less than ¾ inch.

Table LXIX.—Particulars of Screw Propellers in various Ships.

Reference.	Description of ship.	Results of trials.				Particulars of screw.				Ratios of		
		Speed knots.	Revs. pr. min.	I.H.P. / Revs.	D.⅔ S³ / I.H.P.	Diamr. ft.	Pitch ft.	No. of Blades.	Surface sq. ft.	Pitch / Diamr.	Surface. / πr^2	Slip pr. cent.
Mr	Single Screw Passenger S	15·42	76·2	55·52	245	19·00	21·00	4	100	1·105	0·353	2·38
Oo	Do. do. do. S	16·16	64·6	86·66	245	20·00	29·00	4	100	1·450	0·318	12·60
Wdg	Twin do. Express S	22·80	77·2	194·30	281	22·31	32·83	4	...	1·471	...	8·39
Ffs	Single. do. Cargo S	11·39	69·9	52·00	251	19·25	19·25	4	120	1·000	0·412	14·30
Fso	Do. do. Passenger S	14·62	79·75	71·40	246	19·50	21·00	4	118	1·077	0·394	11 65
Dn	Do. do. do. S	16·82	72·00	64·72	270	19·50	26·50	4	115	1·359	0·385	10·70
Ao	Do. do. do. S	14·85	79·00	32·91	220	16·00	20·00	4	85·0	1·250	0·423	4·8
Tns	Do. do. Cargo S	11·27	69·00	28·03	275	16·33	18·50	4	82·0	1·133	0·391	10·55
Fos	Twin do. Passenger S	16·60	89·00	58·18	275	18·75	20·5	4	84·0	1·093	0·304	7·80
Fsf	Do. do. do. S	17·33	89·50	42·15	258	16·50	22·0	4	69·0	1·333	0·322	10·90
Fsn	Do. do. Cargo S	13·56	83·50	35·21	296	16·75	18·25	4	78·0	1·090	0·355	13·56
Fns	Do. do. do. S	12·88	83·00	27·32	220	15·50	17·50	4	61·5	1·130	0·324	10·25
Fso	Do. do. Passenger S	15·09	83·60	25·06	239	14·50	20·00	4	58·0	1·380	0·351	15·09
By	Single do. Cargo S	13·30	83·0	18·69	206	14·25	18·00	4	63·0	1·263	0·396	6·80
Lr	Do. do. do. S	13·23	91·5	17·32	235	13·75	17·00	4	57·0	1·236	0 385	13·80
Eo	Do. do. Passenger S	14·80	88·0	23·66	192	13·50	21·00	4	67·0	1·414	0·469	18·80
Va	Do. do. do. S	14·90	106·0	14·80	224	12·00	16·00	4	48·0	1·333	0·425	11·00
Jo	Do. do. do. S	13·28	101·0	11·81	205	11·50	14·50	4	50·0	1·261	0·481	8·00
Ba	Do. do. Cargo S	10·80	87·0	7·60	228	10·50	15·75	4	36·5	1·500	0·420	15·00
Po	Do. do. do. S	11·30	98·0	5·05	178	9·50	12·75	4	34·5	1·342	0·486	8·30
Qn	Do. do. do S	11·07	132·0	2·46	168	7·75	9·25	4	25·0	1·194	0·530	8·13
Mpl	Twin do. Tug .	11·10	141·0	1·51	123	6·50	10·75	3	17·0	1·654	0·515	25·70
Gtn	Single do. Yacht .	9·96	217·0	0·417	154	4·50	6·75	3	5·5	1·500	0·344	22·0

Table LXIXa.—Particulars of Some Bronze Screws on Fast Ships.

Reference.	Description of Ship and Machinery.	No. of Screws.	Results of Trials.				Particulars of Screws.				Ratios of		Slip. per cent.
			Speed knots.	Revs. per min.	H.P. / Revs.	$D^2/_3 \times S^3$ / H.P.	Diamr. ft.	Pitch ft.	No. of Blades.	Surface. sq. ft.	Pitch / Diamr.	Surface. / πr^2	
Dk	1st Class Cruiser Recipro.	2	23·05	116·0	131·7	234	19 0	24·5	3	76·0	1·29	0·269	17·8
D	,, ,, ,, .	2	24·11	122·4	127·5	262	19·0	23·0	3	105·0	1·21	0·370	13·2
Wr	,, ,, ,, .	2	23·28	139·4	87·5	298	15·75	20·7	3	85·0	1·317	0·436	15·7
Qe	Battleship Direct Turbs.	4	25·0	305·0	60·0	190	12·0	11·20	4	80·0	0·933	0·708	24·2
Lnd	,, Recipros. .	2	18·6	110·0	72·7	246	17·5	19·75	4	86·0	1·129	0·354	13·3
Dgn	Light Cruiser S-Grd. Turbs.	2	29·0	275·0	77·0	165	11·75	14·25	3	82·0	1·213	0·756	25·0
Cld	,, ,, ,, ,, .	2	29·0	330	64·0	147	11·25	11·25	3	75·0	1·000	0·750	20·6
Sct	Squadrn. Ldr. ,, ,, .	2	35·5	355	60·0	138	10·50	13·25	3	72·5	1·262	0·837	18·2
Mst	T. B. Destroyer Direct Turbs	2	37·5	690	24·0	161	8·00	7·50	3	42·8	0·933	0·856	28·4
Dv	T. B. Destroyer Recipro.	2	30·1	386	7·0	239	6·75	9·00	3	16·0	1·332	0·447	12·2
Ptn	,, ,, Direct Turbs.	3	35·4	794	12·1	161	6·33	6·20	3	25·0	0·979	0·781	28·5
Ltna	Ocean Express Direct Turbs.	4	26·5	191·5	94·5	268	16·5	16·13	4	110·0	0·979	0·514	13·2
Olmp	Ocean Express Recipros.	3	22·5	78·3	255	279	22·8	36·0	4	165·0	1·579	0·402	19·0
Tcna	,, ,, S.-Grd. Turbs.	2	17·6	137·0	40·0	299	15·25	15·0	4	73·6	0·984	0·402	12·6
Orps	Passenger Cargo. Dbl. Grd. Turbs.	2	14·5	95·0	40·0	340	17·25	17·5	4	75·0	1·017	0·321	11·8
Sf	Passenger Cargo Recipros.	2	17·0	95·0	48·0	350	17·0	21·13	4	80·0	1·243	0·352	14·1
Dny	,, Mail S. G. Turbs.	2	23·1	287	54·0	212	10·0	10·50	4	43·8	1·050	0·558	22·2
Arh	,, ,, ,, ,, .	2	19·6	254	9·3	238	8·50	9·50	3	30 2	1·118	0·532	17·9
Ko	Cross Chnl. S.-Grd. Turbs.	2	21·0	300	14·0	233	8·70	8·75	3	31·5	1·006	0·533	18·8
Nrm	,, ,, ,, ,,	2	19·7	300	8·3	232	8·00	7·50	3	20·4	0·938	0·408	15·3
Pg	,, ,, Recipros. .	2	20·3	177	19·8	194	10·50	14·3	3	36·0	1·362	0·439	18·9
Pe	,, ,, ,, . .	2	18·9	189	8·5	246	9·00	12·0	3	26·0	1·333	0·409	15·6
Lwc	Yacht S.-Grd. Turbs. .	2	15·3	215	4·2	215	8·00	8·42	3	19·0	1 053	0·380	12·9

THE PADDLE-WHEEL PROPELLER.

The common radial wheel with fixed floats is now seldom seen, inasmuch as it is far less efficient especially with limited diameter than one with feathering-floats. It is, however, a much simpler structure and easy of repair with simple means, and can be constructed largely of timber, as it is indeed on the large tug steamers still employed on the big American rivers, with repair shops few and far between.

There are two positions for this propeller, viz. one wheel on each side, or a single one or pair of wheels at the stern of the ship. The latter position is preferred for river service and in narrow channels. The side wheel is still a favourite propeller for tugs in harbour duties, where manœuvring ability and short, smart service is highly desirable and much backing and filling has to be done.

The effective diameter of a radial wheel is somewhat difficult to determine,—as it depends very much on the form of float, the amount of immersion, the waves set up by the wheel. &c.,—but it is usually measured between the centres of the opposite floats.

Taking the feathering wheel as the one in general use, of diameter D to float centres, moving at R revolutions (at full speed) per min.

A is the area of one float in square feet.

V is its velocity in feet per second and therefore $= \pi D \times \frac{R}{60}$.

S the speed of the ship in knots.

v the velocity of the ship in feet per second $= 1{\cdot}689 \times S$.

f the fraction that the slip is of $V = \frac{V - v}{V}$.

e is the efficiency of the engines.

E the efficiency of engines and wheels.

T_R the tow rope resistance of the ship in lbs. at speed S.

The power transmitted to the wheels is then $= I.H.P. \times e$.

The power delivered by the wheels $= I.H.P. \times E$.

The stream projected by each wheel is measured by $A \times V \times 64$ lbs.

The mass of such stream is $\frac{A \times V \times 64}{32}$ or $2A \times V$.

The acceleration given is $V - v$. Hence

The pressure of float $= 2A \times V(V - v)$ which is equal the thrust.

The work done $= 2A \times V(V - v)v$ or $120A \times V(V - v)v$ per min , which will equal $T_R \times 60V$ and also $I.H.P. \times E$.

Rule 197. The diameter of the wheel $= 19 \times \frac{V}{R}$.

Taking the acceleration as $fV = f \times \frac{\pi D \times R}{60}$.

Rule 198. Then the thrust $= \frac{Af}{1800}(\pi D \times R)^2$. Hence

Rule 199. Area of float $A = \frac{\text{Thrust} \times 1800}{f(\pi D \times R)^3} = \frac{\text{Thrust} \times 182}{f(D \times R)^3}$.

Modifying this rule in accordance with good modern practice, the following rule holds good :—

Rule 199a. Area of one feathering float $= \frac{\text{I.H.P.}}{f - f^2} \times \left(\frac{C}{D \times R}\right)^3$.

If I.H.P. is the gross amount developed and E is $= 0{\cdot}6$, then—

Value of C is 83·2 for a pair of wheels.
Value of C is 109 for a single stern wheel.

In some modern high-class swift steamers E is sometimes as high as 0·66 when clean and in smooth water, in such cases C may be 85·6. On the other hand, for tugs and small ocean-going craft it will be less than 83·2.

The apparent slip of a paddle wheel is larger with a radial than a feathering wheel, and the amount will depend on the circumstances of each case ; where a small "race" only is possible its velocity—that is, its slip—must be greater than if the section were larger. In shallow draught steamers, and on side wheelers for narrow passages through locks or dock entrances, the area of float is necessarily circumscribed—consequently for high power the acceleration must be great. In practice the rate of slip is from 15 to 30 per cent. with radial floats, and 15 to 20 with feathering ; in some cases, due to circumstances above-named, it is even with the latter 25 per cent. at full speed. Generally with a reduction in power and speed of the ship there is a reduction in the rate of slip.

The wheel race of a paddle steamer is not affected nearly so much by hull currents as is the screw, except of course with stern wheels, when wake currents and augmented resistance play a prominent part in propulsive efficiency.

The number of floats on a paddle wheel varies with the diameter.

Rule 200. Radial fixed floats, one for each foot of diameter.

Rule 200a. Number of feathering floats $= \frac{D+2}{2}$ or $\frac{60}{\sqrt{R}}$.

Proportion of Paddle Floats.—A the area, B the breadth ; r the ratio of length to breadth, so that $A = B \times rB = rB^2$; then

Rule 201. **Breadth of float** $= \sqrt{\frac{A}{r}}$.

Rule 201a. **Length of float** $= r\sqrt{\frac{A}{r}}$.

In practice r is 4 to 5 with radial wheels and 2·6 to 3·0 with feathering wheels.

Thickness of floats, when of elm or other equally strong wood, is about $\frac{B}{12}$. Radial floats of common wood, $\frac{B}{8}$ is the thickness. Steel floats stiffened by curvature or angle bars are now often used instead of wood, as having less edge resistance and being cheaper. Their thickness should be 0·16 inch per foot of breadth + 0·16 inch.

N.B.—The edges, both of entry and emergence, should be chamfered at back, and the corners are better rounded.

Formerly wheels were made with two sets of rims, one pair being at the extremity of the arms outside the floats and the other pair just within the floats; in a modern wheel, with the big pitch of float possible with high rate of revolution and to reduce weight, the outer rims are omitted and the arms made as shown in Fig. 43.

Immersion of floats should not be less than one-eighth their breadth, and for general service should be one-half. If the voyage is so long that the fuel consumption materially affects the draught of water, or there is large variation due to other weight carried, the immersion at light draught should be not less than 0·1 × B, and when fully loaded not more than 0·7 × B. That is, for good running the variation in draught ought not to be more than 0·6B, which would mean 2·5 feet for a large ship and one foot for a small one.

To design a feathering wheel so that the floats shall enter edgeways when going at full speed, take P (Fig. 42, page 218), a point on the face of the float just entering the water, draw PA parallel to the water line, and cut off PA to represent the speed of the ship through the water, to some convenient scale; draw PB tangent to the circle through P, whose centre is the centre of the wheel, and cut off PB to represent the speed of the wheel on that circle; complete the parallelogram, and the resultant PR is the direction in which the float enters the water.

Produce RP, and draw parallel to it, at a distance from it equal to distance from gudgeon centre to face of float, a line cutting the circle of gudgeon centres in G.

Draw GH at right angles to PF, set off the breadth of the float, and mark off GL equal to the length of lever required.

Now draw another float whose face is vertical and immediately under the centre of the wheel, and the end of whose lever is M; and with centres M and L, and radius GC, draw two arcs of circles intersecting in E. Then E is the centre of the eccentric pin, and LE the length of the radius rods; and the ends of all the other float levers will lie in a circle struck from E as centre, with radius LE.

Paddle-wheel frames.—These must be strong enough, in every case, to transmit the power of the engines to the water, and must also be stiff enough to work without undue springing or vibration. As the power of the engines is approximately proportional to the cube of the diameter of the shaft, the strength of the wheel frames should be in the same proportion, and the section of the arm would be given by the equation,—

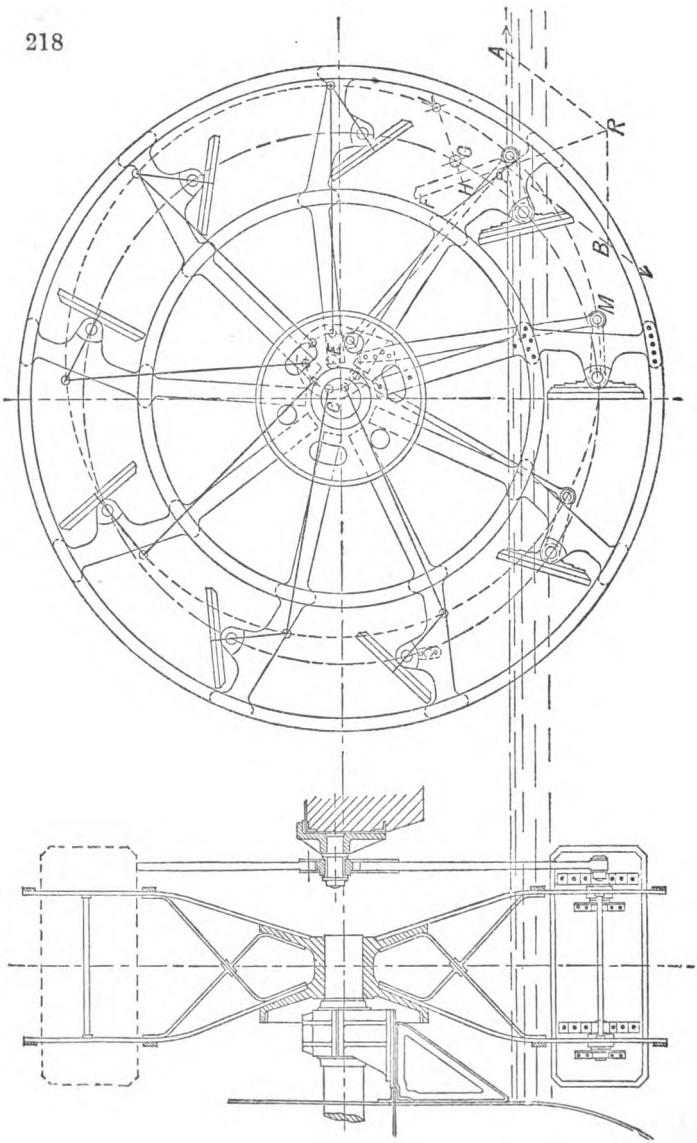


FIG. 42.

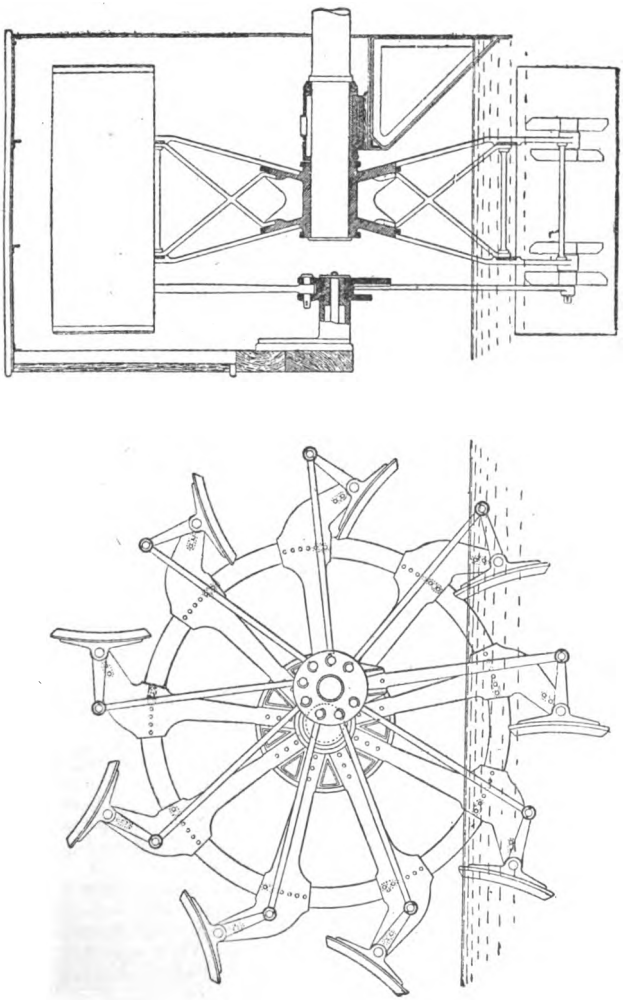


FIG. 43.—Paddle-Wheel with Steel Floats and no Outer Rim.

Rule 202. $$t \times b^2 = \frac{K}{n} \times d^3,$$

where t and b are respectively thickness and breadth of arm, n total number of arms in one wheel, d diameter of inner, or engine, journal of paddle-shaft, and K a co-efficient.

But when the diameter of shaft is small, this gives a section of arm which would not be stiff enough for practical purposes without more cross-bracing than is usual in small wheels; in practice, small wheels are commonly made without any diagonal bracing at all,—the arms being made proportionately stronger,—as experience has shown that it is not advisable to use sections below a certain degree of lightness.

The formula, therefore, requires the addition of a constant quantity, and, if written as follows, gives very satisfactory results:—

Rule 202a. $$t \times b^2 = \frac{K}{n} \times (d^3 + 600).$$

The value of K, for section of arm near boss, is 0·32,—and the other symbols are as defined above.

The formula may be used exactly as it stands for ordinary oscillating engines with sectional crank-shafts (Class 3, Table L.a), but the shafts of the other types are made larger, on account of the greater bending stresses to which they are subject, and what may be called the "equivalent cube of diameter" must be obtained before Rule 201 can be applied. The equivalent d^3 is found by taking the cube of the actual diameter of the journal, and, if for example the engine be of Class 2 (Table L.a), taking $\frac{50}{58}$ths of it.

Theoretically, the arms may be made lighter as they recede from the boss, but in practice it was usual to make them parallel, chiefly for convenience of manufacture, and to use rolled bar.

When there is no outer rim, or ring, the portion of each arm beyond the inner rim must be strong enough to carry its own load without any help from the other arms, and must be considered as a beam fixed at one end and loaded at the other. As there is a tendency to contrary flexure of the arm within the ring, the arm is generally made widest at the ring,—tapering gradually towards the boss, and quickly towards the gudgeon centre or end. The type of wheel is expensive, and the floats are deprived of such protection as is given by the outer rims, and are therefore so much the more liable to injury.

The ratio of breadth of arm to thickness is usually about 5 to 1. The arms must, of course, be locally widened out where the connection to the rim, or rims, is made. When there are two rims, from three to five bolts may be used at each joint, but with one rim only there should be six to ten bolts, according to size of wheel.

Nothing less than a ⅞-inch bolt should be used on ordinary

wheels, and both heads and nuts should be square; also, all holes should be reamered out, and all bolts be driving fit.

Rule 203.

The section of inner ring may be 0·7 × section of arm at boss.

Rule 203a.

The section of outer ,, 0·6 × ,, ,,

Rule 203b.

The section of only one ,, 1·0 × ,, ,,

The proportions of breadth to thickness may be,—

Rule 204. Inner ring . . $\frac{b}{t}=5$.

Rule 204a. Outer ring . . $\frac{b}{t}=4$.

Rule 204b. Only ring . . $\frac{b}{t}=6{\cdot}5$.

In large wheels, the inner rims (or arms near inner rims) should be stayed back to the opposite sides of the boss; the section of stays may be,—

Rule 205.

Section of diagonal stay (if rectangular) ·7 × section of inner rim.

Rule 205a.

Section of diagonal stay (if circular) ·5 × ,, ,,

Rule 206.—Horizontal round ties, or distance rods, should also be fitted between each pair of gudgeon bearings, and as close as possible to them; they may have a diameter of 1·3 × thickness of arm.

It is a common practice to fix the centres of the gudgeons, on which the floats are hinged, slightly off the centre lines of the floats,—nearer the outer edges,—as the strains on the feathering gear are thereby reduced; the amount is to a great extent optional with the designer, but varies in practice from one-tenth to one-twentieth of the breadth of the float.

The whole of the gudgeons and pins should be cased with brass, and work in lignum-vitæ bushes; unless the vessel is to work in sandy or muddy water,—when iron pins working in white-metal bushes will give better results.

The outer bearing for the paddle-shaft may have a length of 1·5 to 2 × diameter of journal,—according to nature of service and consequent weight of wheel; it should be strongly made and firmly fixed, as, in addition to the weight of wheel, the whole of the thrust

is taken by it. The magnitude and direction of the resultant force on the bearing can be easily calculated, but, as its direction is always below the horizontal, the caps do not need to be very substantial.

The lubricant was often mainly water, but oiling pipes and tallow boxes should be fitted, and water kept out.

A stuffing-box should be fitted round the shaft where it passes through the ship's skin.

MARINE STEAM TURBINES.

The turbine is now employed on shipboard to drive the propellers in various ways. In warships it is practically the only motor now in use, and in the mercantile marine it continues to take the place of the reciprocating engine inasmuch as by means of gearing, single and double, the rate of revolution of the propeller can be reduced to permit of the employment of screws of large diameter and consequent high efficiency, especially in a seaway. It is a very cheap form of motor and simple in construction ; its weight is less than that of a reciprocator, and, while occupying no more floor space, it requires much less head room. As a rule it consumes less steam per power unit than the best of reciprocators, and it can make the fullest use of the high vacuum, so cheaply obtainable on shipboard. It can use steam both of high pressure and high temperature due to superheat, with some advantage, but these are not conditions necessary to it for economic or power results, inasmuch as the velocity obtainable with comparatively low initial pressures of steam is sufficient for good results.

The steam turbine is not such an elastic instrument for the marine engineer as the vertical reciprocator, inasmuch as by itself it is not reversible ; at low revolutions it is not so efficient and economic as at high, where the peripheral speed is in accordance with the velocity of flow of the steam ; the speed of revolution even at full speed is not generally so high as it should be for maximum efficiency, and the speed, such as it is, does not permit of an efficient screw propeller. That is, the turbine rate of revolution cannot be fixed arbitrarily by the designer as can be that of a reciprocator. If the rate of revolution is kept low, as it is on shipboard compared with what obtains on shore with turbines driving dynamos, there must be either an increase in diameter of the rotor so as to obtain the peripheral velocity ; or there must be a larger number of stages whereby the "drop" and consequent velocity of flow are reduced ; or there may be, as there generally was, on shipboard both these features, with the corresponding disadvantages of length of instrument and friction losses at the blades and channels.

The solution of these difficulties was attained by the introduction of gearing whereby the rate of revolution of turbine may be as high as

desired or found practicable (now 3,000 to 4,000), while the screw may revolve at the rate for high efficiency under the service conditions. Double gearing is now in general use for large units, and likely to be so for even small ones, as the *spur* or main driving-wheel can thereby be of quite a moderate diameter.

The reversing of the propellers on a turbine-driven ship is accomplished by means of a separate turbine, either in a separate casing, etc., or housed in the rear of the ahead-going motor casing. The provision of power for this purpose is in Germany required to be 50 per cent. of the full power, with a consequent loss of efficiency of the ahead-going turbine of 3 to 5 per cent. ; in England the astern-going power is not so great, being generally about 25 to 30 per cent., when the loss is about 1 per cent. There are now other methods of applying the turbine to propel ships, such as the electric and the hydraulic, whereby the reversal of the propeller is attained without a separate motor. The efficiency of such installations varies from 0·865 of small powers (150 S.H.P.) to 0·900 of the large power in express steamers. So that the losses due to their mechanism are more rather than less those of the astern-going turbine.

The turbines in use in warships and express steamers are the *Parsons*, the *Curtis*, the *Zoelly*, and the *Rateau*. Their efficiency, as judged by steam consumption per power unit on shore, is as high as 70 per cent., but generally it is somewhat less on shipboard. Their consumption of steam at full speed per S.H.P. hour, when of large size, is as low as 12 lbs. under conditions with which the theoretical maximum output of a lb. of steam is 7·64 H.P. (*vide* Table V.).

$$\text{That is, Efficiency} = \frac{60}{12} \div 7{\cdot}64 = 0{\cdot}654.$$

The arrangement of turbines in ships is now as follows :—

(1) A complete head-going machine with all its stages coupled direct to a propeller shaft, and having an astern-going member in the rear of it, both contained in a common casing.

(2) The turbine is divided into two parts, the high-pressure or boiler end is in a separate casing and coupled to its own screw shaft direct; the low-pressure or condenser half has its own casing and is coupled to its own screw shaft, while within the casing is the astern-going turbine. In some cases there is an astern-going turbine in an independent casing, etc., in rear of the high-pressure half.

(3) There is a high-pressure portion in its own independent casing coupled to its own propeller shaft, while there are *two* low-pressure portions, each having its own casing with a reversing turbine, and each coupled to its own propeller shaft.

(4) Each propeller shaft is fitted with a pair of helical steel wheels, side by side, right- and left-handed, and on each side of this

composite spur wheel is a pinion and shaft, one driven by the high-pressure half and the other by the low-pressure half of a compound turbine, as in No. 2.

(5) Each propeller shaft is fitted, etc., as in No. 4, but the rotor shaft is not coupled direct to the spindle which has the pair of pinions, but has interposed a spindle with a pair of pinions which gear into a spur wheel on that spindle.

(6) One or more complete turbines are coupled either direct or by means of gearing to a dynamo, the electricity developed by which is delivered to a motor or motors fitted to the screw shafts direct.

(7) One or more complete turbines are caused to drive some form of pump which delivers its water to a hydraulic motor connected direct to the screw shafts.

In every case the low-pressure member exhausts to an independent condenser, with air, circulating pumps, etc. Direct drive is now a thing of the past with turbines; single gearing involves very large spur wheels to reduce the rate of revolution of the screw to practical limits, and when that rate is that best for large ships, double gearing is a necessity and is now the general practice. The older ships with direct drives followed Nos. 1, 2 and 3 methods.

No. 1 method was the usual one with Destroyers, Scouts, and small ships of very high speed. No. 6 is much favoured in the U.S.A. Navy, while No. 7 was adopted by the Germans in 1913.

The low-pressure portion of the turbine is by far the most effective, and its efficiency is considerably higher than that of a reciprocating engine; on the other hand, the high-pressure portion is not so efficient as the reciprocator. The turbine can also develop a considerable amount of work from the steam as it exhausts from the low-pressure cylinder of a triple compound engine when working at full power, as in it steam can expand to within half a pound of that in the condenser, and that may be with a turbine as low as half a pound (or 29 ins. vacuum) with advantage. Under these circumstances a low-pressure turbine may be attached to a triple compound engine with a considerable increase in power occurring, or the same aggregate power as developed by the triple compound engine may be obtained from the combination with less steam. The gain is usually 15 to 20 per cent. in shore installations. On shipboard this arrangement is easily effected with three screws; there may be, as in the R.M.S. *Olympic*, two reciprocators driving the wing screws, with a turbine taking the exhaust steam from the two reciprocators and driving the centre screw. Or there may be one centre reciprocator and two wing turbines; this is, of course, not so suitable an arrangement except as a means of further compounding a single-screw steamer. The economic result of such a combination can be very good, as the comparative inefficiency of the high pressure end of the turbine is replaced by the more efficient triple reciprocator, and

the comparatively inefficient low-pressure end of what would be a quadruple reciprocator is replaced by the highly efficient low-pressure turbine.

The arrangement of two reciprocaters with twin screws exhausting to a low-pressure turbine driving the centre screw is in every way a convenient one for both merchant ships and war vessels, for the reciprocators are reversible, and so all manœuvring can be done by them with the turbine out of action, and likewise cruising can be done economically. When full speed is desired the exhausts can be easily and quickly switched to the turbine. As a matter of fact, with the reciprocators out of action quite a decent "slow speed" can be maintained by turning the exhaust from all the auxiliaries to the turbine; if supplemented with a modicum of steam direct from the boilers, a better speed can be maintained when required.

This, which may be called the eighth method of applying the turbine system to ship propulsion, is now a popular one, and being used more especially for passenger steamers of moderately high speed working where fuel is dear. The s.s. *Otaki*, of 6857 I.H.P., built by Denny Bros., Dumbarton, having the combination, showed a gain on fuel consumption of 17 per cent. over a sister ship with triple compound reciprocators only.

The steam consumption at full power was 11·95 lbs. per I.H.P. hour. The efficiency under the circumstances was $\frac{60}{11{\cdot}95} \div 7{\cdot}9 = 0{\cdot}635$.

The steam consumption on the main turbine engines of R.M.S. *Lusitania* was 12·77 lbs. per S.H.P. hour. The efficiency in her case was $\frac{60}{12{\cdot}77} \div 7{\cdot}65 = 0{\cdot}614$.

Turbines geared to the Propeller Shafts is the method now in favour as the means whereby the turbine may be used for cargo ships, and the trials made by Sir C. Parsons with the s.s. *Vespasian* amply demonstrated the practicability of it for such a purpose. Since then further experiments with cargo steamers have confirmed the opinions formed by Sir C. Parsons and his friends. But the most interesting development had been the fitting of a cross-channel express steamer with twin-screws driven by turbines geared, and trying her against similar three-screw boats, direct-driven turbine boats, on the same station. The following were the results: the higher efficiency of the geared installation is no doubt due to the fact that by such means the turbines can be run at rates of revolution approaching those on electric generating plants on shore, while the screws can be of such a size and rate of revolution as to give high efficiency of propeller. The efficiency of the *Normania* machinery as measured by steam consumption is $\frac{60}{12} \div 7{\cdot}60$ or 0·658, as against $\frac{60}{15{\cdot}1} \div 7{\cdot}60$ or 0·526 of the *Sarnia*.

Professor Biles states that only 1½ per cent. of the power was lost in

the gearing. That is, its efficiency was 0·985. This is, of course, very satisfactory, and so long as no abrasion takes place it will continue to be so. The wheel and pinions are machine cut, with great accuracy, and the latter run in an oil bath.

Table LXX.—Performances of Channel Steamers with Reciprocators and Turbines.

Particulars of Ship and Machinery.	Twin S.S. P.A. Recipros.	Twin S.S. Chfd. Recipros.	Three S.S. *Sarnia*, Direct Turbines.	Twin S.S. *Normania*, Geared Turbs.
Length between perpendiculars . . . . ft.	290	300	284	290
Breadth . . . . ,,	38·0	34·5	39	36·0
Draught of water . . ,,	11·93	13·8	12·0	12·0
Displacement . . . tons	2,100	2,200	1,990	1,875
Block co-efficient of fineness .	0·581	0·559	0·524	0·524
Wetted skin . . sq. ft.	12,400	13,000	11,900	11,700
Engines	4 Crank Triple	3 Crank Triple	Direct Turbines	Geared Turbs.
Total heating surface. sq. ft.	11,450	11,450	12,985	10,221
Boiler pressure . . . lbs.	180	160	160	160
Propellers, diameter of . ft.	10·5	12·0	5·50	8·0
,, pitch . . ft.	14·5	16·0	4·52	7·12
Revolutions of engines . .	164	130·4	500	1984 & 1380
,, ,, screws . .	164	130·4	500	310
Horse-power developed . .	5,892	4,398	6,670	5,000
Speed, knots	19·12	18·20	20·0	19·7
Displacement$^{2/3}$ × Speed^3 ÷ I.H.P.	194	232	190	232
I.H.P. per 100 ft. of wetted skin	47·5	33·8	56·1	42·8
,, ,, ,, reduced to 10 knots	6·79	5·60	7·01	5·61
Steam consumed per H.P. hour . . . . lbs.	16·75	16·30	15·1	12·0
Coal consumed per H.P. hour ,,	...	...	1·71	1·43

The rate of revolution of a marine turbine can, within limits, be determined arbitrarily, for there is no question of periodicity to complicate matters. There is, however, propeller efficiency to consider, both theoretically and practically, and it is here that the designer finds his limits. He may choose a screw of small diameter with very large surface ratio and quite a good pitch ratio, or he may, for practical reasons, adopt one of larger diameter, when the pitch ratio will be smaller and the surface, as well as surface ratio, smaller; both suitable

to the revolutions fixed on for the turbines. As a rule, the screw of a turbine ship is a compromise between these conditions. If a diameter, such as one would choose for a reciprocator, were used, the pitch ratio would be smaller than consistent with efficiency, and the surface small too—so small that the actual slip would be *less* than that given by revs. × pitch,—the apparent slip would be as high as that observable in a twin-screw ship driven by one screw only.

The actual slip ratio may be, of course, as high as desired by making the diameter small, so that the column of water is of small section—in practice, however, such screws would be inefficient in sea service and be bad for manœuvring.

The following rule may be observed for arriving at a suitable diameter for actual sea service in direct-drive turbine ships.

Rule 208. **Diameter of screw** $= \sqrt{\frac{\text{S.H.P.}}{x - x^2}} \times \left(\frac{\text{C}}{\text{P} \times \text{R}}\right)^3$.

Here P is the pitch in feet; R the revolutions per minute; x is the *real* slip ratio and may be taken as $= 0{\cdot}2\ \text{P}_c + f$, P_c being the prismatic co-efficient of the ship's immersed body, and f is the apparent slip. The value of C is 450. The centre screw, especially of small ships, may be of larger diameter based on $x = 0{\cdot}18\ \text{P}_c + f$.

The following, however, is a simpler rule and may be used with confidence:—

Rule 208a. **Diameter of screw** $= z \times \text{P}_c \sqrt[3]{\frac{\text{S.H.P.}}{\text{R}}}$.

Generally $z = 5{\cdot}75$ for twin and $6{\cdot}55$ for centre screws (when three).

For large ocean-going steamers, $z = 6{\cdot}5$ for all three or four screws.

For T.B. destroyers, $z = 5{\cdot}4$.

The slip ratio in common practice is 13 to 15 per cent. with large ocean-going expresses, and 20 to 22·5 per cent. with smaller cross-channel ships.

The pitch ratio with turbines direct driven is very seldom over 1·0, and generally under; 0·9 to 0·95 is considered good practice.

The actual rate of revolution will be found from these fundamental conditions, and in practice it varied from 185 of the Atlantic expresses to 750 of the cross-channel ones. The *Lusitania* and *Mauretania* did over 190 revolutions at full power with their original screws, and 180 to 185 with new screws of larger pitch ratio.

In the Mercantile Marine, very high-power ocean expresses 185 revolutions, smaller ones 200. Large cross-channel expresses 450 to 500 revolutions, smaller ones 600 to 750, and small excursion and river steamers 750 to 1000.

The tendency latterly had been to reduce the number of revolutions and increase the diameter of screw.

With geared installations it is possible to have a rate of propeller revolution approximating to that which obtained with reciprocators. Our largest and most powerful warships, with an aggregate power

over 100,000 S.H.P., have a rate of about 340 r.p.m. with a turbine velocity of about 3000 revolutions. The large express steamers are run at a rate of about 130 r.p.m., while with the ordinary cargo or cargo-passenger steamer it is as low as 80 to 100 r.p.m. On the cross channel or small express steamer the general practice is 250 to 300 r.p.m.

The diameter of the shafting is, of course, small for the total power, but is based on the power per revolution, which is small. The rotor shaft has to be designed to suit the weight of the rotor and to give the necessary bearing surface. The low-pressure rotor of the *Olympic* weighs complete 132 tons, and has journals 33 inches in diameter; its diameter is 195 inches, and length over all 47·6 feet. That of the *Lusitania* was 184 inches in diameter over all, and its weight 120 tons; it was carried on journals 33⅛ inches diameter and 56½ inches effective length, so that the pressure per square inch was 72 lbs. only, while the speed of rubbing surface was 1650 feet per minute. The pressure on the journals of the high-pressure rotor was 80 lbs., but the rubbing speed only 1350 feet per minute.

Rule 209. Working pressure per square inch of journal $=3000 \div \sqrt{S+100}$ may be taken as a guide in determining the length of journals. Their diameter must be calculated from considerations of torque transmitted, bending moment due to weight, and, to some extent, to inertia, and always with regard to stiffness and the critical speed.

The rules for shafting of turbines are as follows:—

Rule 210. Diameter of Intermediate Shafts $=\sqrt[3]{\frac{\text{S.H.P.}}{\text{R}}} \times \text{F}$.

The value of F generally may be taken as 1,860 ÷ ultimate tensile in tons.

That is with ordinary mild steel 28/32 tons tensile .	F=62
,, ,, high tensile 38/42 tons tensile .	F=46
For light swift Naval ships special steel . . .	F=32
For larger ,, ,, ,, . . .	F=50 to 60
By Board of Trade and the Registration Societies' Rule generally	F=64
By Board of Trade and the Registration Societies' smooth water service	F=58

The diameter of spur wheel shaft, and of the thrust shaft near the collars = diameter of intermediate shaft × 1·05.

If there is only one pinion, or if two pinions whose axes are at angle of less than 120°, the multiplier is 1·1 between the bearings of the axle at spur wheel.

Rule 211. Diameter of Tail Shaft = diameter of intermediate $+\frac{\text{P}}{\text{K}}$, where P is the diameter of screw in inches and the value of K as on page 137.

The rotor drums are generally of forged steel, but sometimes steel castings. They must be most carefully and accurately balanced, especially when the latter is used. The arms or discs which connect the drums to the axle must be stiff enough to prevent springing or distortion, and consequently, then strong enough to transmit the torque. The revolutions of direct driven marine turbines is so low compared with those on shore work, that centrifugal forces and stresses produced were moderate, but still large enough to require attention. With the single and double reduction gearing turbine revolutions are as high as any now made.

The blades of both rotors and stators are usually of bronze, one of the strong zinc-bronzes capable of being rolled, drawn out, or extruded, of the required section, being preferred. These alloys, however, while being strong against tensile forces and soft enough to stand manipulation, are weak against shearing forces; phosphor-bronze is therefore preferred by some engineers, as this can be drawn into strips, while possessing a hardness and high elastic limit superior to the zinc-bronzes. It will also stand better the eroding action of steam at high velocity. Steel blades are also fitted, especially where sufficient section cannot be got in the ordinary way to permit of the bronze. Steel alloys—especially the nickel ones—have been used. Corrosion,* however, is not altogether unknown—due often to leaky condenser or other similar defects admitting sea-water to the boilers. Too much care cannot be devoted to the workmanship when blading the rotors, and the same may be said of the stators, but the latter are not subject to such severe trials as the former. Each builder of turbines has his own design of blade, choice of material, and method of fixing. The strip system is simple, inexpensive, and quite effective when properly done; at high rates of revolution, with corresponding high centrifugal action on them, they are not always so trustworthy; in the past, with direct coupled turbines on shipboard, this has not had to be dwelt on, now, however, with geared ones at 3000 revolutions, the same attention must be given to it as obtains on shore. Since high vacuum steadily and cheaply maintained is essential to the economic working, it is absolutely necessary there should be no air leaks anywhere, and the low-pressure turbine journals should be provided with means to prevent air passing through them when they are leaky; water or steam passing are of no consequence, so provision may be made for one of them to seal the journal.

The design of a turbine is really a simpler problem than that of a compound reciprocator; it is, however, started in quite a different way, except that in both cases the intended rate of revolution is assumed. With the turbine it is usual to begin with a rate of steam consumption per unit per hour; with an initial pressure of 175 lbs. absolute, 15 lbs. is a covering allowance with a vacuum of 28 inches. At higher initial pressures the available work is, of course, greater; but with them the turbine is not so efficient.

* The special stainless or non-corrodable steels are now often used, and with care are quite satisfactory.

Quantity of steam per second per 1000 S.H.P. $= \frac{1000 \times 15}{60 \times 60}$, or 4·17 lbs.

Of this, from 3 to 5 per cent. will disappear by leakage, so that the nett amount through first guide blades will be only about 4·0 lbs., or 10·28 cubic feet. If the flow there is 250 feet per second, and the rate of revolution is 480 per minute or 8 per second,

Area through guide blades $= \frac{10{\cdot}28 \times 144}{250} = 5{\cdot}92$ square inches, and so

on throughout the various stages during which the steam is expanding, acquiring velocity, and giving out its kinetic energy to the rotor blades.

The loss of area through the annulus, in which the blades act due to the blade section, is so considerable that it may be taken that:—

The gross area $= 5{\cdot}92 \div 0{\cdot}32$ per 1000 S.H.P., or 19 nearly.

Rule 213.

$$\text{Mean diameter of rotor in inches} = \frac{\text{Velocity of steam} \times 12 \times 60 \times f}{\text{Revs.} \times \pi}.$$

Factor f is the ratio of the velocity of the blades to that of the steam, and is usually 0·4.

In this particular case,

$$\text{Mean diameter} = \frac{250 \times 12 \times 60}{480 \times 3{\cdot}1416} \times 0{\cdot}4 = 47{\cdot}7 \text{ inches.}$$

Height of blade $= 19 \div \pi \times 47{\cdot}7$, or 0·127 in. per 1000 S.H.P. In practice, such short blades would not be fitted; instead, a considerable portion of the first ring on the stator would be blocked, leaving ports in which the blades would act.

Rule 214. **The size of exhaust passage** is, of course, very great, for, if it is assumed that the flow to condenser is at a pressure of 1½ lbs., the volume will be about 900 cubic feet, which, with a rate of flow of 160 feet per second, the area of section will be 810 square inches per 1000 S.H.P.

Rule 215. The length of blades should never exceed 15 per cent. of the diameter of rotor; from 8 to 10 per cent. gives the best results. Sometimes they are only about 1 per cent., in which case the leakage is bound to be high, inasmuch as the clearance space is high compared with the blade annulus.

The losses of a marine turbine have been summarised by Mr S. J. Reed as follows:—

Leakage over dummies Friction of bearings, glands, &c. Radiation Loss by carry over of kinetic energy from last row of blades of low-pressure turbine	8 to 12 per cent.

Friction of steam in blade channels Leakage over tops Steam shock and eddying Loss of kinetic energy due to splitting action of blades	35 to 45 per cent. H.P. 25 to 35 per cent. L.P.
Loss of carry over from last row blades of high pressure	

Testing of turbine cases is made by water pressure after being machined, but before blading, &c.

Rule 216. *The Admiralty require* the test pressures to be as follows:—

(1) High-Pressure Turbine, ahead and astern.

Steam end and inlet cover	255 lbs.
The remainder	200 ,,
After completion	170 ,,

(2) Cruising Turbine.

Steam end and inlet cover	255 lbs.
The remainder	200 ,,
After completion	170 ,,

(3) Low-Pressure Turbine.

Astern cylinders and inlet cover	50 lbs.
Ahead cylinders and inlet cover	30 lbs.
Exhaust chambers and passages	30 ,,
After completion	30 ,,

(4) Receiver pipe and valve between cruising and high-pressure ahead turbine . . 255 lbs.

(5) Steam chambers of condenser and eduction pipes . 30 lbs.

Rule 217. *The Board of Trade require* high-pressure turbine cases to be tested to 1·33 × working pressure. Low-pressure cases, condenser end, 30 lbs. The astern going cases to working pressure.

Rule 218. *The Italian Government demands* are somewhat similar, viz.:—

Casing of high-pressure turbine to 1·33 × working pressure; astern going to working pressure; low-pressure turbine, admission end, 0·33 × working pressure, and exhaust end 28·5 lbs. per square inch. Condensers for turbines 28·5 lbs.

The weights of turbine installations are less than that of reciprocators.

Rule 219. Total weight of turbine machinery (Naval) with water-tube boilers

$$= \frac{\text{S.H.P.} \times 16{\cdot}85}{\text{Revs. per minute} - 100} \text{ tons.}$$

Mercantile installations on Board of Trade rules, with cylindrical boilers and full complement of auxiliaries, as in first-class steamers:

Rule 220. Total weight machinery $= \frac{\text{S.H.P.} \times 44}{\text{Revs. per minute} + 50}$ tons.

(i) *Example.*—Cross-channel steamer, 8000 S.H.P., revs. 500.

$$\text{Total weight} = \frac{8000 \times 44}{550} = 640 \text{ tons or } 179{\cdot}2 \text{ lbs. per S.H.P.}$$

(ii) *Example.*—Naval scout of 28,000 S.H.P. at 750 revs.

$$\text{Total weight} = \frac{28{,}000 \times 16{\cdot}85}{750 - 100} = 726 \text{ tons, or } 58{\cdot}1 \text{ lbs. per S.H.P.}$$

The steam consumption of a turbine ought, according to Professor Rateau, to be nearly as given by his formula.

Rule 221.

$$\textbf{Steam consumption} = 2{\cdot}13 + \frac{16{\cdot}2 - 2{\cdot}05 \log_e P}{\log_e P - \log_e p} \text{ lbs. per hour}$$

P being the initial and p the final pressures.

The power developed by and transmitted from a turbine is found by ascertaining the torque or twisting moment from observations of the twist or angular displacement of a definite length of shaft. Then for shaft diameter d :—

If T_1 is the torque in inch-lbs., R the revolutions per minute, θ° the angular displacement in degrees,

$$\text{S.H.P.} = \frac{T_1}{12} \times \frac{2\pi R}{33{,}000} = \frac{T_1 \times R}{63{,}000},$$

$$\text{or} \quad T_1 = \frac{\text{S.H.P.}}{R} \times 63{,}000.$$

$$\text{Now} \quad T_1 = \frac{\theta^\circ \times 20{,}120\, d^4}{L} \text{ for solid shafts, and}$$

$$T_1 = \theta^\circ \times 21{,}147 \left(\frac{d^4 - d_1^4}{L} \right) \text{ for hollow with bore } d_1.$$

It follows on combining these equations that

Rule 222. (1) $\text{S.H.P.} = \frac{R}{L} \times \frac{d^4}{3{\cdot}13}$ for solid.

Rule 222a. (2) $\text{S.H.P.} = \frac{R}{L} \times \frac{d^4 - d_1^4}{2{\cdot}98}$ for hollow shafts.

L is the length of shaft under observation, generally taken as 40 inches to suit the usual torsionmeter.

The formula in every day use is as follows :—

Rule 223. $\text{S.H.P.} = \frac{\theta^\circ d^4 \times R}{Q \times L}$ for solid, and $\frac{\theta^\circ (d^4 - d_1^4) \times R}{Q \times L}$ for hollow shafts, where Q is taken as 3·27, the modulus assumed being 11,250,000, which is for the ordinary mild-steel shafts of commerce. The Admiralty require a somewhat stronger steel (32 tons ult-tensile), and always have the actual shaft tested mechanically with its own torsionmeter in the shops before fitting into the ship, to ascertain the

value of θ° for a series of twisting moments. From these observations a curve for the value of T_1 can be made and the value of the factor determined for each shaft and ship.

The **torsionmeters** in use are either wholly mechanical, like the Föttinger, which can, and does actually register the torque by a continuous line on a sheet of paper like an indicator diagram; or they may be partly mechanical, the registering being made by a beam of light from a mirror actuated by the mechanism thrown on a scale, which can be read off while the shaft is running. Hopkinson and Thring's instrument is of the latter kind, and gives good and reliable results. Then there is the Bevis-Gibson apparatus, where a beam of light is transmitted through slots in discs, two on the shaft and one fixed. These are in line when the shaft is at rest or running without load, and can be brought into line when running under load by the angular displacement of the fixed disc. The amount of displacement necessary is the measure of the twist. There is a fourth kind, in which synchronism is indicated by sound, and the production of this condition when running under load is effected in such a way that the compensation indicates the twist. This latter is the Denny-Johnson.

Trials of Turbine-driven Ships.

The following results of trials with ships fitted with turbines, or reciprocator-turbine arrangements, are interesting and instructive.

(1) R.M.S. *Lusitania*, a four-screw Atlantic express steamer, 760 feet long × 87·5 feet beam, and on a draught of water of 31·6 feet (mean) displaced 35,600 tons (*v.* Table X.).

Steam and Fuel Consumptions of R.M.S. "Lusitania," with Turbines running on Sea-service Conditions at Various Speeds.

Speed of Ship in Knots per Hour.	15·77	18·00	21·00	23·00	25·40
Shaft horse-power	13,400	20,500	33,000	48,000	68,850
Consumption of steam per hour, total . . . . lbs.	284,500	253,600	493,300	668,300	879,500
Consumption of steam auxiliaries lbs.	71,000	76,400	85,700	96,700	116,500
Consumption of auxiliaries per cent. of total	25·0	21·6	17·4	14·51	13·2
Steam consumption per S.H.P. hour, turbines . . . lbs.	21·23	17·24	14·91	13·92	12·77
Steam consumption per S.H.P. hour, auxiliaries . . . lbs.	5·30	3·72	2·60	2·01	1·69
Steam consumption per S.H.P. hour, total . . . . lbs.	26·53	20·96	17·51	15·93	14·46
Coal consumed per S.H.P. per hour, total . . . . lbs.	2·52	2·01	1·68	1·56	1·43
Total consumption of fuel on voyage, 3100 miles . tons.	2980	3190	3670	4520	5390
Time on voyage . . hours	203	178	152	139	126

(2) S.S. *Otaki*, a three-screw passenger and cargo steamer for New Zealand service. She is 464·5 ft. × 60 ft. × 34 ft., and on a mean draught of 27·5 feet displaces 9900 tons. Her trials, however, were carried out on 20 feet mean draught. She was fitted with twin screws, each driven by triple reciprocators having cylinders 24·5 inches, 39 inches, and 58 inches by 39-inch stroke, exhausting into a low-pressure turbine, 90 inches diameter, driving a central screw. The following shows the results of progressive trials and a comparison with the performance of a sister ship, s.s. *Orari*, driven by twin screws, each having cylinders 24·5 inches, 41·3 inches, and 69 inches by 48-inch stroke.

Measured Mile Trials of S.S. "Otaki," with Mixed Machinery, 31st October 1908, on 20 feet Mean Draught.

	Mean of A Runs.	Mean of B Runs.	Mean of C Runs.	Mean of D Runs.
Total horse-power, being I.H.P. (reciprocators) + S.H.P. (turbine)	6857	5348	4704	3282
Mean speed . . . . . knots	15·02	14·28	13·83	12·52
Revolutions, reciprocators	103·5	97·9	93·5	83·4
,, turbine	224·5	209·7	197·2	172·1
Total water consumption per hour . lbs.	82,000	67,300	60,200	44,600
Total water consumption per hour, per H.P. . . . . lbs.	11·95	12·6	12·8	13·6
Mean absolute pressure at H.P. cylinder lbs.	193	178	166	135
,, ,, turbine inlet lbs.	9·5	7·62	6·76	5·0
Vacuum at exhaust end of turbine	28·1	28·2	28·4	28·5
,, on condenser gauge	28·2	28·4	28·3	28·5
Temperature of sea-water . . . F.°	56	56	56	56
,, circulating discharge	70	67	70	70
,, hot well	72	70	73	74
Steam consumption based on the I.H.P. of s.s. *Orari* and measured by tanks	13·66	13·7	13·8	13·07
As measured by pumps per I.H.P. per hour	14·12	14·1	14·3	15·2

Name of Ship.	E.H.P.	I.H.P.	Propulsive Co-efficient.	Water Consumption per Hour.		
				Total.	Per E.H.P.	Per I.H.P.
			Per cent.			
3-screw s.s. *Otaki* (turbo-reciprocators)	3350	5880	57	73,300	21·9	13·7
2-screw s.s. *Orari* (reciprocators)	3210	5360	60	88,800	27·5	16·5
Gain per cent. in *Otaki*.	..	..	..	17	20	17

(3) The two ships compared here are channel steamers of the L. & S.-W. Ry. Co., both driven by turbines (*v.* Table LXX.).

(*a*) S.S. *Sarnia*, a three-screw steamer, is 284 ft. × 39 ft. × 23·8 ft. moulded, and on a mean draught of water of 12 feet displaces 1990 tons. She has three screws, driven direct by the usual installation of high-pressure and low-pressure turbines running at about 500 revolutions per minute. Her speed, with 6670 S.H.P. developed, is 20 knots, giving an Admiralty co-efficient of 190 only.

(*b*) The S.S. *Normania*, a twin-screw ship, is 290 ft. × 36 ft. × 23·5 ft. moulded and a displacement of 1900 tons on 12 feet draught. She has two screws, each driven by two turbines geared by pinion and spur wheels to the propeller shafts, and maintained a speed of 19·7 knots with 5000 S.H.P., showing a co-efficient of 232·4, or 22·5 per cent. higher than the *Sarnia*. The fuel consumption per voyage of the *Normania* is, however, as much as 40 per cent. less than that on the *Sarnia*. This vast difference is due to three causes: (i) the superior efficiency of the turbines, due to the 1984 revolutions per minute of the high pressure and 1380 of the low pressure; (ii) the superior efficiency of screws, 8 feet diameter, running at 300 revolutions, over those of only 5·5 feet diameter at 500 revolutions; and (iii) to the better proportion and form of hull with the twin screws. The total steam consumption of the *Sarnia* was as much as 17·1 lbs. per S.H.P. hour, whereas the *Normania* used only 14·3 lbs., or 16·4 per cent. less.

(4) The trials of H.M.S. Cruisers *Amethyst*, driven by turbines and three screws, and the *Topaze*, by twin screws and triple reciprocators, are interesting, but hardly fair to the latter, as her engines had cylinders, etc., for great power without regard to economy of consumption; so that it was only at reduced speed that economic conditions existed. As showing the variation of the two systems the following is perhaps more interesting.

Water Consumption per I.H.P. per Hour of H.M.S. "Amethyst" and "Topaze" on Progressive Trials.

Speed in Knots.	10	11	12	13	14	15	16	17	18	19	20	21	22
H.M.S. *Amethyst* (turbines) .	29·3	26 0	23·7	22·0	20·4	19·0	17·9	16·8	15·9	15·2	14·7	14·3	14·0
H.M.S. *Topaze* (reciprocators) .	23·8	22·0	20·6	19·6	18·8	18·4	18·3	18·4	18·7	19·2	19·8	20·5	21·4

(5) The series of trials of the Cruisers, "City" Class, is instructive. These ships are alike and 430 ft. × 47 ft. × 26·75 ft., having a displacement

of 4800 tons and 15·25 feet mean draught. The *Bristol* has Curtis turbines and twin screws, the others the ordinary Parsons turbines with four screws. Each ship has twelve Yarrow water-tube boilers. The following are the results of the various trials:—

Trials of the Quadruple Screw Cruisers, "City" Class.

	H.M.S. *Glasgow.*	H.M.S. *Liverpool.*	H.M.S. *Newcastle.*	H.M.S. *Gloucester.*	H.M.S. *Bristol.**
A. 22 hours at 66⅔% full power.					
S.H. Power . . .	14,055	14,102	14,051	13,968	14,300
Revolutions per minute .	430	426·5	420·5	..	..
Mean speed, knots . .	23·70	23·88	23·34	23·45	24·06
Coal per S.H.P. hour .	1·61 lbs.	1·57 lbs.	1·65 lbs.	1·59 lbs.	..
B. 8 hours at 80% power.					
S.H. Power . . .	18,220	18,824	19,116	18,983	19,130
Revolutions per minute .	471	464	461·6	..	..
Mean speed, knots . .	24·90	25·10	24·84	25·08	25·17
Coal per S.H.P. hour .	1·57 lbs.	1·59 lbs.	1·65 lbs.	1·48 lbs.	..
C. 8 hours at full power.					
S.H. Power . . .	22,500	24,718	25,417	24,340	24,230
Revolutions per minute .	499	512·7	518·2	..	..
Mean speed, knots . .	25·80	26·17	26·27	26·31	26·84
Fuel per S.H.P. hour .	1·63 coal	1·65 coal	1·036 coal ·171 oil	1·14 coal ·42 oil	
			1·207	1·56	..
D.⅔×S.^{3}÷S.H.P. . .	217·3	211·5	201·3	211·3	227·3

* H.M.S. *Bristol* has twin screws.

(6) S.S. *Reina Victoria Eugenia* is a four-screw passenger-cargo steamer, 480 ft. × 61 ft. × 35·8 ft. Of the four screws, the outer pair are each driven by a separate low-pressure turbine, 95 inches diameter, taking steam from the corresponding reciprocator triple driving one of the inner pair of screws, and having cylinders 29 inches and 43 inches, and two low-pressure 47 inches diameter by 42 inches stroke. There were two crucial trials, one deep laden to 24·7 feet mean draught, and the other on 19·85 feet, as she would be loaded with light cargo or partly laden with heavy goods.

Gain per c.

Trials of 4-S.S. "Reina Victoria Eugenia."

	Fully Laden.	Part Laden.
Displacement . . . . . tons	13,229	10,181
Horse-power developed . . . I.H.P.	5,760	7,340
S.H.P.	2,157	3,500
total H.P.	7,917	10,840
Revolutions per minute . . { reciprocators	102·9	112·6
turbines	395·0	481·0
Speed mean . . . . . knots	16·10	18·12
Slip per cent. . . propellers of recipros.	4·0	1·0
,, turbines	13·0	20·0
Admiralty co-efficient $D^{\frac{2}{3}} \times S^3 \div I.H.P.$ . .	294	257
Consumption of steam per I.H.P. hour . total	12·375	15·17
,, ,, ,, only main engines	10·625	...
Steam pressure at valve chests of recipros. lbs.	170·50	170·00
,, ,, turbine entrance . . lbs.	5·40	7·80
,, ,, condenser . . . lbs.	0·56	0·50
Temperature of sea-water	45° F.	43° F.
,, discharge water . . .	64° F.	70° F.
,, hot well	62° F.	70° F.

Steam was supplied by seven single-ended boilers, 16 feet 3 inches diameter × 11 feet 6 inches long, having a total heating surface of 20,965 square feet so that 5·67 lbs. of steam per hour were given by each square foot of T.H.Surf., with an air pressure of 0·35 inch in the ash-pits with Howden's system of draught.

(7) S.S. *Cairncross* is 370 ft. × 51 ft. × 27·8 ft., of 9950 tons displacement on 23·4 feet mean draught, the block coefficient 0·779. She has a single screw of large diameter, driven by two (high-pressure and low-pressure) turbines geared by pinions and spur wheel. Steam is supplied from three boilers at a pressure of 180 lbs. Her normal speed in fine weather is 10 knots.

Her sister ship, the *Cairngowan*, has similar boilers and propellers, but driven by triple compound engines, with cylinders 24 inches, 40 inches, and 66 inches diameter and 45 inches stroke. They were laden practically to the same displacement, supplied with the same coal, and steamed side by side for 36 hours. The weather, however, was bad, so that no account is made of the speed.

It will be seen that the turbine proved a more economic instrument than the reciprocator, and was worked with a vacuum of 28·75 inches

as against 26·8 inches in the *Cairngowan*. No doubt, too, that in rough seas the turbine-driven screw did not race so much as the other, and thereby got drier steam, etc.

The ratio of spur wheel to pinions is 26.

Special 36 Hours' Sea Trials of S.S.'s "Cairncross" and "Cairngowan."

	S.S. Cairncross.	*S.S. Cairngowan.*
Machinery, kind of	Geared Turbines	Triple Compound
Mean revolutions of screw, p.m.	61·76	61·68
Horse-power developed	1570 S.H.P.	1790 I.H.P.
Steam pressure at engines	138 lbs.	..
Temperature of hot well . . F.°	79°	104°
Vacuum in condenser . . ins.	28·75	26·80
Coal consumed per 24 hours . tons	27·8	32·7
,, ,, ,, I.H.P. hour lbs.	=1·45	1·704
Coal per sq. ft. of grate . . ,,	17·9	21·0
Water per hour . . . . ,,	22,400	27,200
,, ,, I.H.P. hour . . ,,	=12·57	15·18
Ash from coals . . per cent.	8·50	8·97
Assuming calorific value 14,500 B.T.U., the efficiency of system is	12·10	10·32
Saving of fuel . . . per cent.	15·0	..

Actual Steam Consumption on Two Large Liners.

Description of Machinery.	S.H.P.	Steam at Turbines.	Consumption in Lbs. per Hour.		
			Main Turbines.	Auxiliaries and other Purposes.	Total for all Purposes.
4 Shafts direct-coupled turbines . .	23,000	Saturated	11·2	2·2	13·4
2 Shafts geared compound turbines, 1 H.P. and 1 L.P. to each.	14,500	85° F. superheat.	9·7	2·3	12·0

Methods of Transmission from Turbine to Screw Compared. (Cleghorn.)

Points of Comparison in a Ship of 20,000 S.H.P.	Method of Transmission.		
	Double Wheel Gearing.	Hydraulic Transformer.	Geared Electrical Generator, and Motor Driven.
Rev. of turbines per min. .	2400 and 1500	1000 (about)	2400 and 1500
,, screws ,, .	90	140	90
Transmission efficiency . .	0·965	0·92	0·885
Relative fuel consumption	100	123	109
,, machinery weights	100	109	108

INTERNAL COMBUSTION ENGINES.

These engines are such as consume their fuel in the same cylinder or other vessel in which the energy is imparted to the mechanical means of motion. The fuel may be in the gaseous state produced from coal, oil, or other volatile substances giving off hydrocarbon, or it may be the spray from oils injected into the cylinder.

Gas engines on board ship have not proved an unqualified success, and since they are generally supplied from producers, which are heavy and occupy much space, they are at a considerable disadvantage in competition with oil engines.

Oil engines may be divided into three classes:—

(*a*) Those using petrol or other *light* volatile oil.

(*b*) Those working with paraffin, having a higher flash point.

(*c*) Those working with heavy oils, having such a high flash point as to be perfectly safe on shipboard.

Petrol, so much in demand for locomotion on land, is too dangerous and now too dear for maritime purpose generally. It is convenient, on account of its vapourising so readily, for starting engines whose general fuel is heavy oil. With a flash point of 70° F. it is also used on small pleasure boats, where very high speed is required. It is too dangerous for handling in confined engine-rooms or on shipboard at all.

Paraffin, which is cheaper than petrol, can be got anywhere, has a flash point of 90° to 120° F., which renders it safe to handle, and, as may be seen by Table LXXI., it has high thermal value, quite as high as petrol. Paraffin engines are largely employed on small craft for propulsion with success, and the demand for them, since petrol is becoming so dear, is likely to increase. The flash point of paraffin is, however, too low for the Admiralty, whose limit is 175° F., as it is also for Lloyd's, whose limit is 150° F.

Alcohol, which can be produced so cheaply everywhere, has not yet been used seriously for propulsion inasmuch as the flash point is high and uncertain, so that with a Diesel or Semi-Diesel engine it misses fire often and cannot be relied on. Its calorific value is much lower than that of the petroleum products, being only 10,260 B.T.U. per pound of 90 per cent. mixture, or not much more than half that of good heavy oil.

Heavy oils, with flash points from 175° to 250°, are largely used now on shipboard as well as on shore. They are mostly residuals of kinds, even when called crude oils; in fact, unless an oil has undergone some sort of a refining process, it is not fit for an oil engine to use. Some residuals are left after all the lighting and lubricating oils have been abstracted. Some are crude oils after some exposure to the air to get rid of sulphur and dangerous volatile elements. In some so-called crude oils there remains an amount of bituminous matter which gets converted into hard sandy coke; this asphaltum really spoils the oil for internal combustion purposes, as the coke formed from it will not fuse or consume away, but remains in a crystalline state fit to cut the pistons, valves, &c.

Table LXXI.—Oils and other Liquids used as Fuel in Internal Combustion Engines.

Name of Liquid.	Specific Gravity.	Flash Point.	Boiling Point.	Thermal Value per lb.	Composition.	
					Hydrogen.	Carbon.
		F.°	F.°	B.T.U.		
Light Oils in general	0·65 to 0·76	70	300	..	..	..
Gasoline	0·66 to 0·67	..	158 to 176	..	..	..
Benzine	0·68 to 0·75	..	176 to 302	..	..	..
Petrol in general	0·66 to 0·72	..	158 to 230	18,360	14 to 16	86 to 84
Heavy Oil, medium	0·76 to 0·86	70 to 158	300	20,000 to 18,000	13·6 to 14·3	86·3 to 82
Paraffin, Russian and American	,,	,,	,,	19,000	..	..
,, Russian (best)	0·82 to 0·83	90	..	19,000	..	..
Standard Oil	0·81	84	..	19,000	..	..
Heavy Oils in general	0·86 to 0·94	140 to 270	..	18,800	12·3 to 13·7	86·6 to 83·5
Solar Oil	0·86 to 0·89	140 to 270	..	17,400 to 18,600	..	..
Texas Oil	0·91 to 0·95	130 to 270	..	18,000 to 18,900	10·9	84·6
Galician Solar Oil	0·86 to 0·88	.	..	18,500	12·6	85·3
Borneo Oils	0·91 to 0·95	176	..	17,600	..	..
Alcohol Methyl	0·834	..	..	10,300	12	43 to 45
,, and Benzol, mixed	..	..	..	13,700	..	..
,, American, denatured	0·8199	..	..	10,512	12·68	47·04
Tar Oil, distillate	0·92	..	..	17,500	..	..
Benzol	0·882 to ·885	..	175	17,100	..	..
Shale Oil, Swedish	0·997	130	..	17,487	9	87·8
,, French	0·911	..	..	18,700	11·5	80·3
,, Scotch	0·86	220	..	16,775	7·0	86·5
Creosote Oil	..	400	..	15,800	7·6	82
Mexican Crude	0·950	over 150	..	18,800	11·68	83·52

N.B.—The British Admiralty require that the oil fuel admitted on board His Majesty's ships shall have a flash point no lower than 175° F., and shall contain no more than 0·75 per cent. of sulphur.

The weight of air required for the combustion of petrol and paraffin oil is 184 times that of the oil; for heavy oils, as used on Diesel engines and the residuals in boilers, 176 times is sufficient. Benzol requires only 152, and alcohol (90 per cent.) will do with 96 times. Petrol generally has a flash point over 80° F. and Paraffin over 120° F., as supplied for use in engines.

Creosote, or tar oil, is a home product which can be used in these engines quite well, especially after a little primitive refining; the calorific value is 15,800 B.T.U. per pound.

Shale oil, as distilled from the shale in Scotland, is also a valuable one for fuel and easier to run with than the petroleum residuals; it is freed from naphtha spirit and paraffin wax, and then has a calorific value of 16,775 B.T.U. and a flash point of 220° F.

In course of time other liquid fuels will be found which may be used in oil engines, for already the late Dr Diesel had tried successfully some vegetable oils.

Petroleum, as found in various parts of the world, differs somewhat in chemical composition and characteristics; they are sometimes classed as *Asphaltic* and *Paraffin* oils in America; the Russian and East Indian oils can hardly be so classed, however.

Asphaltic oils are of high specific gravity (0·945), high in sulphur, harden under atmospheric influences, and their residues tend to solidify; they are subject to decomposition at comparatively low temperatures, and their distillates have a high solvent power.

Paraffin oils are of lower specific gravity (0·832), are low in sulphur, are not affected by exposure to air, have good capillary power, and their distillates have only low solvent properties.

The sulphur in the Californian (Asphaltic) oils is as much as 3·3 per cent., as against only 0·3 per cent. in Texas oil. The British Admiralty used to require their oil supplies to have no more than 0·75 per cent. of sulphur and the flash point to be not lower than 200° F.; now 3 per cent. of sulphur and 175° are permitted. The boiling point of such oil is about 300° F.

Oil engines are subject to another classification, viz:—

(1) Those which draw in the charge of fuel with the air and compress the mixed charge lightly, say to 80 lbs. pressure per square inch, before explosion.

(2) Those which draw in a charge of air only and compress it moderately, say to about 150 lbs., or about 60 per cent. of the pressure produced by the explosion of a charge of oil vapour sprayed on to a hot plate or sphere. This kind is known as the *Semi-Diesel.*

(3) Those which draw in a charge of air and compress it so highly (400 to 500 lbs.) that it becomes sufficiently hot to ignite the charge of oil spray on entering it and produce an explosion without shock.

This system, wherein the air is compressed by itself alone to a temperature sufficient to ignite the charge of oil spray and to burn it without any serious increase to the pressure, was invented and perfected by the late Dr Diesel, and is known as the Diesel System.

There is still another differentiation among oil engines, and now, for marine engineers, an important one.

Both the *Diesel* and the *Semi-Diesel* oil engines may be worked on either the two-stroke or the four-stroke cycle. The latter is the older

and known as the "Otto"; most gas and land oil engines work on it, and altogether it is simple and reliable; but it involves larger or more numerous cylinders for the same power, consequently for marine propulsion the two-stroke cycle is the one in general use.

In the Otto cycle, with a single-acting engine, the piston makes an outward stroke, and thereby sucks into the cylinder a mixture of gas and air in the ordinary engine and a charge of air only in the Diesel engine. On the second, or return stroke, there is compression more or less, as already named; ignition takes place, so that this, the third, is the active stroke; on the fourth, or return stroke, the products of combustion contained in the cylinder are driven out, and it is called the scavenging stroke.

In the two-stroke cycle the cylinder is scavenged and filled with fresh air, so that the first stroke of the piston compresses it; at the commencement of the second stroke fuel is sprayed in, ignited, and burnt during that stroke; at the end fresh air is again caused to scavenge and charge. In this way there is an active stroke every revolution, while in the Otto there is one only for every two revolutions.

The double-acting oil engine, as the name signifies, is active on both sides of the piston, and can, as a consequence, have smaller cylinders still. So far, however, this kind of engine has been successful only in small high-speed craft, but it is said the German Naval ships of large size are being so fitted. The Augsburg-Nürnberg Co., however, have made, and are still engaged in experiment with and perfecting of, engines of this kind of considerable size for Naval purposes.

For obvious reasons, with such very high initial pressures and high temperatures, cylinders of very large diameter are almost impossible and certainly undesirable. Consequently, for large power, a large number is usually found.

The number of cylinders in marine oil engines, for this cause and as a consequence of the difficulty of keeping the engine running steadily at low revolutions, must be numerous; and even when there are four cylinders a heavy fly-wheel is really necessary for safe running "dead slow." There are seldom fewer than four cylinders to marine oil engines, and often double that number when the power is large, so that in twin-screw ships there are sixteen cylinders, and even in quite small ships as many as twelve. The four-cycle engine, single acting, should, of course, never have less than four cylinders on a single line of shaft, whatever the power may be, and even then a fly-wheel is necessary for steady running.

In a general way it may be assumed that, whatever type of oil engine is used, the larger the number of cylinders the better it runs; the torque is more uniform, and the certainty of continuous running at "dead slow" is ensured. On the other hand, the larger the number the more numerous are the working parts all requiring attention, lubrication, and power for driving.

No marine oil engine of the two-cycle type should have less than three cylinders, even if double acting; and all single-acting ones should have not less than four. For an even torque six cylinders are best, and then the fly-wheel can be dispensed with.

For very large powers six and eight cylinders must be employed on each line of shafting. It is convenient in such cases to double the engine into two sets of three or of four cylinders, each capable of running independently, so that with a clutch or other good coupling between them it can be arranged to run only one half when for cruising or other purpose low power only is required.

It should be noted that the efficiency of the two-stroke cycle is 2 per cent. less than that of the four-stroke, for a separate compressor gives a loss of 4 per cent.

Although the fuel of the Diesel and Semi-Diesel is a safe heavy oil, these engines will not start with it, so that a small stock of a more volatile oil is necessary for this purpose, as well as to clean out with just before stopping if some hours are to elapse before restarting.

The paraffin engine also requires, as a rule, a charge or two of petrol or ether to start with.

Any oil engine will work readily ahead or astern if it can be started into motion and ignition caused to take place just as the piston commences the down stroke. The starting and manœuvring of such engines, which proved difficult in the past with even small sizes, are now effected easily and quickly by means of compressed air.

The reversing of the propeller has been accomplished in small craft by such means as turning the blades through a considerable angle with the engine running, or by means of wheel gearing, as in the motor car.

In the large engine of to-day there is always the means of starting and reversing provided, whereby air compressed to very high pressure is admitted to the oil cylinders by means of valves operated by gear which is reversible, as in a steam engine; and it is claimed that the large oil engine in the hands of trained engineers is as handy and reliable for manœuvring as any steam engine.

The general design of the marine oil engine varies somewhat; in the small sizes there is usually the enclosed casing surmounting the bed-plate and supporting the cylinders, besides providing guides, etc., as introduced by Belliss & Morcom for their high-speed electric generating engines with forced lubrication, &c. The larger engines are designed generally on the lines of the marine vertical steam engine, with such modifications as are necessary for oil-engine appliances.

The cylinders have a piston stroke longer in proportion to the diameter than usual with steam; their "heads" and covers have to be very strongly constructed and secured to resist shock; in design and material generally the greatest care is necessary, having regard to the high temperature as well as high pressure to which they are exposed. Water-jacketing is resorted to freely, and seeing that 40 per cent. of the heat is carried away by the cooling water, the circulation must be free.

The pistons are always fitted with Ramsbottom spring rings freely distributed to prevent leakage at all times. Water-cooling is resorted to here in big engines, and in double-acting engines of all sizes cooled pistons are necessary. In the two-cycle engine the scavenging pump is often immediately below the piston, so that a piston-rod is some-

times necessary to make the connection with the connecting-rod. Generally speaking, a piston-rod is necessary and always desirable to keep the connecting-rod top end in a cool place. It is certainly not desirable to have it in a trunk close to the piston face.

The guides and shoes may be just as in a steam engine and follow the same rules for surface, but if the initial pressures are taken for purposes of calculation a higher nominal pressure per square inch can be adopted.

Rule 224. Gross area of guide shoe $= \frac{T \times \sqrt{S + 100}}{3050}$ sq. ins.

T = area piston × initial pressure ÷ 5,
S = piston speed in feet per minute.

The crankshafts are subject to considerable straining from the intermittent action of the pistons as well as the great range of pressure —the maximum being over 5 times the mean.

Shafts of Oil Engines.—Lloyd's Rules when shafts are made of ordinary mild steel. If special steels are used, the size must be submitted for consideration.

Rule 225. 1. Diameter of crankshaft in inches $= C\sqrt[3]{D^2 \times S}$.

D is the diameter of cylinder and S the stroke of piston, each in inches. C is a multiple of value as follows for smooth-water service :—

Four-Stroke Cycle.	Two-Stroke Cycle.	Bearing between each Crank.	Two Cranks between the Bearings.
For 1, 2, 3, or 4 Cylinders	1 or 2 Cylinders	C = 0·34	C = 0·38
,, 6 ,,	3 ,,	C = 0·36	C = 0·40
,, 8 ,,	4 ,,	C = 0·38	C = 0·425
,, 12 ,,	6 ,,	C = 0·44	C = 0·49

For the Open Sea Service add 0·02 to each value.

Rule 226. Diameter of screw shafts in ins. $= G\sqrt[3]{D^2 \times S(N + 3)}$.

N is the number of cylinders ; if two-stroke, N = twice the number. The following are the values of G :—

	Smooth Water Service.	Open Sea Service.
No ma intermediate shafts, . . .	G = 0·155	0·165
three cyli shafts with continuous liners, .	G = 0·170	0·180
should have ,, separate or no liners,	G = 0·180	0·190
best, and the		

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With large thrust collars the shaft between them must be $^{21}/_{20}$th the diameter of shaft elsewhere. For Diesel engines and others with high initial pressures, sizes must be submitted.

Rules of the Bureau Veritas for the shafting of Internal Combustion Engines.

Rule 227. Diameter crankshaft in ins. $= 0{\cdot}106 \sqrt[3]{G \times D^2 \times S}$.

D = diameter of cylinder in inches,
S = stroke of piston ,,
G = a variable factor.

Values of G for a Ratio $\frac{S}{D}$.

$\frac{S}{D}$	Four-Cycle Single-Acting Engines.				
	6 Cylinders.	8 Cylinders.	10 Cylinders.	12 Cylinders.	16 Cylinders.
	Two-Cycle Single-Acting Engines.				
	3 Cylinders.	4 Cylinders.	5 Cylinders.	6 Cylinders.	8 Cylinders.
	Two-Cycle Double-Acting Engines.				
		2 Cylinders.		3 Cylinders.	4 Cylinders.
1·0	127·867	129·849	135·082	139·046	151·657
1·1	119·224	121·336	126·894	130·989	144·184
1·2	112·221	114·431	120·281	124·506	138·156
1·3	106·425	108·700	114·745	119·165	133·140
1·4	101·526	103·931	110·139	114·714	129·021
1·5	99·421	99·891	106·261	111·331	125·501
1·6	93·863	96·365	102·943	107·708	122·528
1·8	88·139	90·739	97·564	102·439	117·714
2·0	83·693	86·422	93·430	98·480	114·015

Rule 228. The diameter of the intermediate shafts, $d_1 = n\sqrt[4]{D^2 \times S}$. The values of the factor n are given in the following table :—

Four-Cycle Single-Acting.	Two-Cycle Single-Acting.	Two-Cycle Double-Acting.	n.
6 Cycles	3 Cycles	..	0·389
8 ,,	4 ,,	2 Cycles	0·397
10 ,,	5 ,,	..	0·415
12 ,,	6 ,,	3 Cycles	0·427
16 ,,	8 ,,	4 ,,	0·459

Rule 228a. The diameter of the thrust shaft, in way of thrust block, $d_2 = d_1 + 6{\cdot}4$ per cent.

Rule 229. The diameter of the propeller shaft, $d_3 = d_1 + \frac{D}{100}$, where D is the diameter of the propeller in inches.

In estimating the fuel consumed per unit of power transmitted for use, full account must be taken of *all fuel* used in auxiliaries; moreover, as these engines require a very large amount of lubricating oil, it would be only fair to include it as fuel in both the oil and steam

engine. In both cases the consumption (total) in pounds per hour should be divided by the Brake Horse-power, or Shaft Horse-power.

The auxiliaries or additions required by the two-stroke cycle marine engines are (i) a scavenging pump delivering air direct to the oil cylinders, (ii) a pump for compressing and storing the air necessary for manœuvring the engines, (iii) another compressor for the air of very high pressure for feeding and spraying the fuel, (iv) a pump to circulate the cooling water; (v) a pump to deal with the fuel.

(i) The scavenging pump is generally an ordinary single-acting air pump worked by the main engine, and having a capacity from 1·25 to 1·35 that of the cylinders supplied.

(ii) The pump for the supply of air for handling the engine is also generally worked by the main engine and delivers at a pressure of about 300 lbs. per square inch, for which purpose it has to be of the two-stage type. This pump, however, has been lately arranged to be worked by an independent engine.

(iii) The air for oil supply is of higher pressure—about 750 lbs.; it is taken from the 300-lb. reservoirs and further compressed by a small pump worked from the main engine.

The fuel consumption of the Diesel engine of the two-stroke cycle type is generally about 0·45 lb. per B.H.P. hour; the mechanical efficiency of these of good size is about 0·78; so that the consumption per I.H.P. is 0·351 lb. Some large engines when new and on trial with skilled hands have done better, viz., 0·38 lb. per B.H.P. or 0·300 lb. per I.H.P. hour. Generally speaking, half a pound of good heavy oil (calorific value 19,000 B.T.U. per lb.) per B.H.P. is a good and satisfactory result in everyday work.

When this is done the efficiency (thermal) is 0·343 (*v.* Rule 17). Tested in this way, the turbine consuming 12 lbs. of steam would require to use 0·70 lb. of equally good fuel and the efficiency would be 0·191. The ordinary turbine with a consumption of 13·5 lbs. would show only 0·167, or less than half that of the oil engine.

The Diesel engine's consumption per B.H.P. at lower powers does not increase so rapidly as does that of steam engines as may be seen by the following table.

Table LXXII.—Consumption of Oil at Various Loads per B.H.P. Hour in lbs.

Load.	400 B.H.P. Engine.		200 B.H.P. Engine.	
	Crude Oil (Sulzer).	Crude Oil (Chalkley).	Creosote Oil.	Crude Oil.
Full power, . .	0·471	0·473	0·518	0·442
Nine-quarters load, .	0·471	0·475	0·462	0·462
three ead, . .	0·477	0·496	0·528	0·506
should rter load, .	0·546	0·493		
best, and				

Vegetable oils are of course not so suitable for internal combustion engines as the petroleum ones, but some can be used with advantage as demonstrated by the following :—

Table LXXIIa.—Vegetable Oils free from Water and Impurities which may be used in Diesel Engines.

	Palm.	Arachis Grand Nut.	Cotton Seed.	Sesame.	Average.
Specific gravity at 15° C. ($H_2O=1000$)	913·8	926·4	923·5	922·2	921
Viscosity Engler at 50° C.	3·47	3·63	3·43	3·37	3·47
,, 100° C.	1·27	1·47	1·29	1·21	1·31
Flash point C.° . .	280	258	243	257	259·5
Combustion tempr. C.° .	325	300	286	299	302·5
Congealing point C.° .	27·42	...	...	...	...
Spontaneous combustion in air	400	400	...	...	...
Moisture per cent. . .	3·10	0·09	0·10	0·08	...
Impurities per cent. .	4·25	Trace	Trace	Trace	...
Calorific value B.T.U. .	16,884	16,924	16,785	16,840	16,858

Composition.

	Palm.	Arachis	Cotton Seed.	Sesame.	Average.
Carbon per cent. . .	76·80	76·60	77·25	76·80	77·1
Hydrogen ,, . .	11·90	12·10	11·70	12·13	11·9
Oxygen ,, . .	10·90	11·00	10·65	10·50	10·6
Sulphur ,, . .	0·009	0·012	0·008	0·010	0·1

N.B.—11 to 12 lbs. of air required per lb. of oil.

The mechanical efficiency of the marine oil engine is not so high as that of a steam engine of equal power, as the friction, both internal and external, is greater than that of a quadruple compound, besides which there are the various pumps driven by the main engine which require more power than the air and feed of the steam one. It is probable that with the large slow-moving engines of the mercantile marine the

mechanical efficiency is under 78 per cent., and all of this class of engine show poor efficiency at lower speeds, so that when cruising the general efficiency will not be so very superior to the reciprocating steam engine.

The mean pressure in the marine oil engine at full power is from 95 to 115 lbs. per square inch. In special cases it is 125 lbs., and efforts are being made to increase this for very high-speed warship engines in Germany. Generally for good and economic working 100 to 110 lbs. is a fair allowance for estimating purposes.

The rate of revolution can be more or less arbitrarily decided on; as a rule, it is higher size for size than prevails in steam engineering in the mercantile marine.

Burmeister & Wain, with their associated friends, have a kind of standard rate which varies from 250 revolutions per minute with single-cylinder 30 H.P. to 150 revolutions for 1000 H.P. four-cylinder engines. All these engines from 40 to 250 H.P. may have two cylinders; from 50 to 750 H.P. they supply also three-cylinder ones; while they make four-cylinder engines of 50 H.P., which they run at 225 revolutions and up to 1000 H.P. at 150 revolutions.

The Augsburg-Nürnberg Co. design their 36 H.P. engines for 250 revolutions with two cylinders, and make four-cylinder 200 H.P. for 195 revolutions, while their 1000 H.P. are for 150 revolutions.

Sulzers will make engines of 8000 H.P. to run at 105 revolutions. Naval engines are designed for higher rates, so that the 1200-H.P. engine is made for 400 revolutions.

The *Selandia* of 2500 I.H.P. has engines running at 140 revolutions; each engine is, of course, 1250 H.P. The *Eavestone* was designed for and has run at 90 revolutions, indicating 1370 I.H.P.

The following table gives the rates of revolution generally aimed at by the Augsburg-Nürnberg Co.

Table LXXIII.—Rates of Revolution of Augsburg-Nürnberg Oil Engines.

Size of Engine.	Revolutions per minute.	
	Light Naval.	Heavy Type.
150 horse-power, . .	550	400
200 ,, . .	550	330
300 to 500 horse-power, .	500	330 to 275
600 horse-power, . .	450	275
900 ,, . .	420	260
1200 ,, . .	400	215

The rate of revolution of marine engines may be determined by the following rules, and be in accordance with the above and other good practice:—

Rule 230. Number of revs. per min. $= 2300 \div \sqrt[4]{\text{H.P.}}$ Light naval.

Rule 230a. ,, ,, ,, $= 2300 \div \sqrt[3]{\text{H.P.}}$ Heavy ,,

Rule 230b. ,, ,, ,, $= 1500 \div \sqrt[3]{\text{H.P.}}$ Mercantile.

The weight of the oil engine is considerable per H.P., so that an oil-engine installation is often as heavy as a steam one.

(*a*) The very lightest, such as made specially for small Naval ships, weigh 40 lbs. per S.H.P. of the smaller and 35 lbs. of the larger powers, as against 43 lbs. of reciprocator destroyers.

(*b*) The ordinary mercantile installation weighs from 300 lbs. of small to 215 lbs. of large engines per S.H.P. The slower speed ones, such as the *Eavestone*, are heavier still.

(*c*) The engines made for express short-passage ships are, of course, considerably lighter, being from 90 to 100 lbs. per I.H.P. The machinery of similar ships with reciprocators would be 210 lbs. per I.H.P., and turbine installations 123 lbs.; while that of a third-class cruiser with water-tube boiler is 120 lbs. with reciprocators, and 95 with turbines.

The space occupied by oil-engine installations is very much less than that by steam ones, and practically by the amount of the boiler room, for even the 8-cylinder engines require little or no more space than the 4-crank engines and condensers of the steamship. The space occupied by oil fuel as against that by coal is a most important factor. Leaving aside the question of taxing the space in the same way as coal bunkers, there remains the fact that oil can be stowed at little cost for labour in spaces which are inaccessible for cargo, and brought from them to the engines and boilers at equally small cost. The application of the fuel requires no hand labour and practically costs nothing, so that altogether the labour costs in an oil-fuel ship are very small indeed. Besides which, the weight of the firemen not required, together with that of the food, water, and appliances for them, which will be considerable, is saved and in its place freight-paying cargo is carried. On the other hand, it must not be forgotten that whereas any kind of fuel can be used in a boiler, only certain kinds and qualities of *oil* can be used in an oil engine; so that in case of accident the oil is lost, a further supply is not to be got, and the oil-engine ship is helpless. Moreover, in this country we have an abundant supply of coal, but we have little or no oil—nor have we as yet an oil-field near at hand in foreign countries. As long as crude oil could be got at 40 to 50 shillings a ton it was worth while having an oil engine here, especially for intermittent work, but the demand before the war had caused it to rise to over 90 shillings; it is now more costly than coal with a steam engine, and likely to be still higher if the mercantile marine goes in largely for this class of engine, as this would expand the demand enormously; but doubtless other oil-fields will be found.

The advantages of the Diesel system over others are:—

(1) Safety due to the use of a harmless high flash-point fuel.

(2) High compression of air free from fuel, whereby:—
(*a*) The use of an ignitor is avoided.
(*b*) Explosion, or combustion rather, is effected without shock.
(*c*) The mixture of fuel with the air is more certain.
(*d*) A poorer mixture can be used when desirable.
(*e*) There is no fear of pre-ignition or back-firing.

(3) Less cooling water is required consequent on the less fuel consumed.

The Semi-Diesel engine possesses most of these advantages, but it requires an ignitor, which is liable to fail in action when dirty.

Of the heat generated by the combustion of the fuel, 38 to 40 per cent. is converted into work, 40 per cent. is absorbed by the cooling water or lost by radiation, and 20 per cent. passes away at the exhaust. A considerable amount of the 60 per cent. otherwise wasted can be recovered by means of a steam boiler, as the temperature at exhaust is about 450° F.—that is, sufficiently high to evaporate some of the water from the jackets which has been raised to near the boiling point.

The steam from this auxiliary boiler can be employed for domestic purposes as also for steering and driving the auxiliary machinery generally.

It means that if in the *Eavestone* 33 per cent. of the loss can be recovered, there should be produced 1000 lbs. of steam per hour; enough to supply auxiliaries of an aggregate of 46 I.H.P.

The indicated horse-power can be found as follows:—

D is the diameter of the cylinders in inches whose number is n;
S is the stroke of piston in feet; p_m is the mean pressure;
R is the number of revolutions per minute.

Rule 231.

$$\text{I.H.P.} = \frac{D^2 \times S \times p_m \times R}{25{,}338} \times n$$

$$\text{Diam. of cylinder} = \sqrt{\frac{\text{I.H.P.} \times 25{,}338}{S \times p_m \times R \times n}}$$

for 4-stroke cycle single-acting.

Rule 232.

$$\text{I.H.P.} = \frac{D^2 \times S \times p_m \times R}{12{,}669} \times n$$

$$\text{Diam. of cylinder} = \sqrt{\frac{\text{I.H.P.} \times 12{,}669}{S \times p_m \times R \times n}}$$

for 2-stroke cycle single-acting.

Rule 233.

$$\text{I.H.P.} = \frac{D^2 \times S \times p_m \times R}{6335} \times n$$

$$\text{Diam. of cylinder} = \sqrt{\frac{\text{I.H.P.} \times 6335}{S \times p_m \times R \times n}}$$

for 2-stroke cycle double-acting.

Limitation to diameter of cylinder is enforced by one or two practical considerations; the thickness will increase with the diameter

and soon become so great as to seriously retard the passage of heat to the cooling water, and the strength of the shell to resist internal pressure and shock will not be proportional to the thickness. The surface exposed to heat increases as the diameter, while the quantity of heat will vary as the square of the diameter, so that the ability to keep sufficiently cool decreases as the diameter increases. In practice with the requirements of the mercantile marine, the diameter seldom exceeds 24 inches, and probably 30 inches is the safe limit with cast-iron; latterly, however, several firms both at home and abroad make engines with cylinders nearly 30 inches diameter and piston stroke 43 to 45 inches. On special short service craft where spurts at high speed are required cylinders of larger diameter may be employed.

The thickness of cylinder is, of course, much greater for merchant ships than for fast expresses and naval craft.

Rule 234. Thickness of cylinder, $= D \times f$ for Diesel engines.

For merchant ships, $f = 0{\cdot}1$ to $0{\cdot}08$.

Naval and expresses, $f = 0{\cdot}06$ } of superior iron.
Very light engines, $f = 0{\cdot}05$ }

The cylinders are often made thicker at the end where there is greatest compression and the explosion; then the thickness tapers and varies from $f = 0{\cdot}10$ to $f = 0{\cdot}06$.

The cylinders of Semi-Diesel engines may be, of course, thinner, so that f is $= 0{\cdot}6$ of the above values of f.

This means that the metal, which must always be of a special kind, is nominally stressed to 2750 lbs. per square inch in mercantile engines, and as much as 5000 lbs. in naval high-speed engines.

The following are a few examples of the performance of the heavy oil engine in actual practice.

(1) *A Diesel engine* with cylinders 22·05 inches diameter × 29·5 inch stroke running at 150 revolutions per minute.

The consumption of oil fuel per hour was 195·2 lbs.

The maximum pressure in cylinders, 525 lbs. per square inch.

The mean ,, ,, 95 ,, ,,

The indicated horse-power—full speed, 609; slow speed, 163·3.

The brake ,, ,, 475·5; ,, 54·6.

Fuel per B.H.P. hour, lbs. ,, 0·411; ,, 0·838.

Efficiency, B.H.P. ÷ I.H.P. ,, 0·78; ,, 0·334.

The pumps were driven by a special engine whose efficiency was 0·715 and the horse-power 44·8. Taking into account the fuel expended on these engines and the B.H.P. of the main engine only, the consumption of fuel was 0·444 full power, 1·415 lbs. low power.

(2) *The S.S. Eavestone* had Diesel engines (Westgarth Carel) with four cylinders 20 inches diameter, 36 inches stroke in which a mean pressure of 127 lbs. has been obtained. She attained a rate of 9 knots, loaded on a consumption of 3·5 tons of crude oil fuel per twenty-four hours, which is equal to 0·47 per S.H.P. hour. For this speed there would be about 900 I.H.P. developed. The mean pressure with 90 revolutions

would be only 83·4 lbs. A sister ship, with modern good triple expansion engines, requires 12½ tons of Durham coal to do the same speed. If it cost 20 shillings per ton *f.o.b.*, the oil to equal it in cost would be at the rate of 71·4 shillings per ton, which is lower than was ruling for residuals. The great saving, however, in favour of the oil engine is in wages, etc., of firemen, which would amount to £20 to £25 per month in a low-power boat like the *Eavestone*.

On the other hand, it is pretty certain that engine-room accounts for overhauls and adjustments at terminal ports will be much heavier in the oil-engine ships; repairs and renewals will also figure largely. Moreover, the energies and attention of the engine-room staff will be more sorely tried, especially on voyages with lots of stoppages.

(3) *The Dorsetshire* of the Bibby Line is a twin-screw ship having oil engines developing on trial 4500 I.H.P. at 115 revs. This ship is 450 ft. long and 57 ft. beam, and 7500 G.R. tons. Speed 12 knots. To each propeller there are six cylinders 29·13 ins. dia. with a piston stroke of 45·28 ins.; they work on the four-stroke cycle and are single acting.

A German ship 525 feet long and of 9800 G.R. tons, has two sets of engines each with six cylinders 22·6 ins. dia. and a piston stroke of 39·4 ins., works on the two-stroke cycle. Another ship on the two-stroke cycle has one engine with six cylinders 23·6 ins. dia. and 43 ins. piston stroke. It develops 2300 I.H.P.

(4) *A Semi-Diesel engine*, having a cylinder 17 ins. dia. × 27·5 ins. stroke, when running at 180 r.p.m. compressed to about 300 lbs. per square inch, and after explosion the pressure was 500 lbs. The thermal efficiency was 40 per cent. and the mechanical 68 per cent., thus the general efficiency was 27·2 average throughout trial.

(5) *A Semi-Diesel engine* by Beardmore has four cylinders, 9 ins. dia. × 13 ins. stroke. It runs at 400 r.p.m., compresses to 150 lbs., and reaches 300 lbs. after explosion. A diagram taken when running on service shows maximum pressure 275 lbs. and mean 52·5, giving the ratio 5·24.

MOTOR BOATS AND OIL ENGINES USING PETROL.

Board of Trade Rules for.

Instructions to Surveyors, 1913.

The following instructions are issued by the Board of Trade for the information and guidance of their officers in surveying motor boats for which a passenger certificate is required.

201. . . . The provisions of the Act as to passenger steamers, namely, section 267 and sections 271 to 288, apply, in so far as they are applicable, to motor boats, and therefore no motor boat may carry more than twelve passengers unless it has been surveyed within the preceding twelve months and holds a passenger certificate which is still in force, and is applicable to the voyage or excursion on which the vessel is about to proceed.

The Surveyor . . . must be satisfied that the hull and machinery are sufficient for the service intended and in good condition; and in considering this point, he must have regard to the conditions of wind and weather to which the vessel may be exposed, and must see that she is so constructed as to be able to meet these conditions without danger to human life.

. . . The Surveyor should be satisfied that proper precautions have been taken to guard against the occurrence of fire or explosion, and that adequate provision has been made for extinguishing fires should they occur.

202. **The duration of the certificate.** It is desirable in many cases to limit it to three months at first, but certificates for a longer period may be issued subsequently, if the engines are found to have worked satisfactorily, if the tank and pipe connections and fittings have proved oil-tight, and if everything in good condition.

203. The printed regulations relating to the survey of passenger steamers apply, as modified and supplemented by the provisions of this circular, sections 201–228.

The following requirements apply to launches or boats fitted with engines using petrol or other petroleum.

204. **Oil Tank.**—The oil tank should be well and substantially constructed of suitable material, and should not be larger than is reasonably necessary. When made of iron or steel, it should be efficiently galvanised externally; if it is proposed to galvanise it on the inside, the work should be carefully done, otherwise the zinc coating may become detached by the corrosive action of the petrol. The tank and its connections should be perfectly oil-tight, and should be tested by hydraulic pressure corresponding to a head of water of at least 15 feet.

Particulars of the tank and its fittings should be submitted, if a pressure feed system is employed.

205. **Tray for Oil Tank.**—The tank should be securely fixed in position, and should rest in a properly supported and suitable lead-lined or metal tray, situated above the deep-load line, with drain pipes from the tray leading overboard.

206. **Arrangements for filling Oil Tank.**—In order to reduce the risk of explosion, the arrangements for filling the tank should be such that oil will not readily be spilled, or overflow and drain into, or lodge in, either the compartment containing the tank or any other part of the vessel; and provision should be made whereby the petrol vapour which is displaced when the tank is replenished will be discharged overboard. If the tank is filled through the deck, the woodwork surrounding the inlet pipe should be covered with sheet metal to prevent it becoming saturated with oil or spirit. A properly secured wire-gauze diaphragm, or tube, which should be made readily removable for cleaning and examination, should be fitted to each inlet and outlet on the tank; and the filling pipe, or orifice, should be furnished with a suitable screwed cap.

No loose cans of petrol should be carried in the boat, and the permanent petrol tank should be charged with petrol when the passengers are not on board.

207. **Safety Device for Oil Tank.**—Means should be provided for relieving the pressure in the tank in case of fire. This may consist of (*a*) an open pipe with wire-gauze diaphragm fitted, (*b*) a light spring-loaded valve, or (*c*) a satisfactory fusible plug or similar safety device.

208. The pipe conveying the petrol to the carburettor should be solid drawn copper and provided with a flexible bend or bends; a cock or valve should be fitted at each end of the pipe one on the tank, and the other on the carburettor or float chamber, and the joints and the couplings should be readily accessible and such that they can be made and kept perfectly oil-tight. In this connection it should be noted that coupling and other joints made of soft solder are not satisfactory.

The air inlet pipe to the carburettor should be fitted with a wire-gauze diaphragm and carried to the ship's side or to a vertical height of at least six feet above the carburettor, and to such a position that there will be no danger of ignition of any petrol vapour that may escape therefrom when the engine is stopped. It is desirable that carburettors should be of such a type that when the motor is stopped the supply of petrol to the carburettor will be shut off automatically. In some cases a suitable receptacle is required to be fitted to the carburettor to prevent an overflow of petrol from the latter into the launch when the engine is stopped; this receptacle should be formed with a narrow neck and provided with a wire-gauze covering at the mouth and with means for draining it.

The exhaust pipe should be efficiently cooled to prevent danger.

209. **Ignition.**—An exposed spark gap should not be permitted in the engine-room, and care should be taken that the leads from the accumulators or generators to the sparking plugs are efficiently insulated and well secured and protected from moisture, particularly when the high-tension system of electrical ignition is adopted.

Ignition tubes should not be passed unless oil having a higher flash point than 73° Fahrenheit is used. If blow lamps are used for this class of oil, they must be fixed and the flame enclosed.

210. **Motor Compartment, Ventilation, &c.**—If the motor, or petrol tank, is situated below deck, it should be confined within a separate watertight and well-ventilated compartment, in which no stove or other apparatus for containing fire should be placed. The compartment should be furnished with at least two cowl ventilators, so arranged as to prevent the accumulation of oil vapour in the lower part of the space, to which part one of the ventilators should extend. Any enclosed space within which the motor, or tank, is placed should be ventilated in this manner, except in the case of small open launches where louvres, or other suitable openings can be provided, in which case one cowl ventilator may be passed if the arrangements are to the Surveyor's satisfaction. In a vessel of this class, the space occupied

by the motor, petrol tank, &c., should, preferably, be at the after end of the boat, and separated from the space allotted for the accommodation of passengers and crew by a substantial bulkhead as high as the seats, and watertight for at least the lower half; but, if it is specially desired to place the motor amidships, or forward, either arrangement may be allowed, provided a bulkhead, formed in the manner stated, is placed between the motor space and the passenger or crew space.

211. **Tray for Motor.**—If the vessel is of wood, a metal tray which can readily be cleaned should be fitted under the motor; and, if flooring boards are provided, they should be closely fitted, but made removable in order to facilitate cleaning and inspection.

212. **Machinery to be fenced where necessary.**—The machinery should be fenced where necessary, in order to prevent injury to the persons in the boat.

213. The foregoing are the more important requirements relating to the construction and arrangement of oil motors, and those which must be complied with generally; but it should be clearly understood that the details of each case will be considered separately by the Board of Trade, and it is only when this has been done that all the requirements which must be fulfilled can be determined. It is therefore necessary that the Surveyors should be in a position to submit to the Board at an early stage full particulars of the case, including plans of the motor, oil-tank, carburettor, and reversing gear, in order that any necessary alterations may be pointed out before the construction of the machinery is commenced, if possible.

To ensure this being done, it is necessary that the owners or builders of motor boats or engines should make early application for survey, when such is required, by filling in the form Surveys 6 and sending it, with the necessary fee, to the nearest mercantile marine office.

In all cases the limits within which the vessel is intended to ply should be stated.

The material of which the various parts are made should be shown on the plans submitted. The steel portions should be tested in the presence of the Surveyor and should be of ductile quality, and the scantlings of all the parts should be sufficient for the purpose intended.

214. **The cylinders** should be satisfactorily tested by water pressure in the Surveyor's presence to double the maximum working pressure to which they will be subjected, and the silencer and exhaust pipes to at least one-fourth of the pressure applied to the cylinders, and they should, owing to the possibility of back-firing, be of ample strength.

215. **The boat should be tried** in the Surveyor's presence for handiness in manœuvring, going ahead, stopping and going astern, before the declaration is issued.

216. **The means for extinguishing fire** to be provided in the case of boats less than 30 feet in length should consist of at least one chemical fluid fire-extinguisher of an efficient size and pattern, and a box of sand of not less than one cubic foot in capacity; a suitable scoop for applying

the sand should also be provided. In larger boats, or in special circumstances, additional fire-extinguishing appliances may be required. The extinguishers should be of sufficient strength to withstand the pressure generated by the combination of the chemicals employed when the outlet valve to the vessel is closed. The extinguishing medium must be of a nature that will not be harmful to anyone with whom it comes in contact. Permanent instructions should be attached to each apparatus showing how it is operated, and it should be stated whether the apparatus may be stowed in any position or in a certain position only. The extinguishers should be placed in a position ready for immediate use, and the plunger for breaking the glass containing the acid should be protected from accidental movement.

It is desirable that petrol or other tanks containing highly volatile and inflammable liquids should be provided with a safety fitting to relieve the internal pressure, and prevent the tank from bursting in the event of its becoming overheated by fire. Whenever such fittings are about to be adopted, particulars should be first reported for consideration.

217. The foregoing instructions apply to all new motor boats and to all motor boats which come under survey for the first time. In the case of boats which have already been passed, the Surveyors need not insist on alterations being made in order to secure full compliance with the regulations, provided they are fully satisfied with the present arrangements; but whenever any repairs or renewals are being effected, the regulations should be fully complied with. This does not apply to the requirements of paragraph 23 as to fire-extinguishing appliances, which must be complied with in all cases.

LLOYD'S RULES FOR INTERNAL COMBUSTION MARINE ENGINES (other than Diesel Type).*

General.

Section 1. In vessels propelled by internal combustion engines, the rules as regards machinery will be the same as those relating to steam engines so far as regards the testing of material used in their construction and the fitting of sea connections, discharge pipes, shafting, stern tubes, and propellers.

Construction.

Section 2. 1. The following points should be observed in connection with the design of the engines.

2. The shaft bearings, connecting-rod brasses, the valve gear, the inlet and exhaust valves must be easily accessible.

3. The reversing gear and clutch must be strongly constructed and easily accessible for examination and adjustment.

4. In engines of above 60 B.H.P. which are not reversible and which are manœuvred by clutch, a governor or other arrangement must be fitted to prevent racing of the engine when declutched.

* For Diesel type, Rules, &c., *vide* Appendix H, p. 695.

5. Efficient positive means of lubrication (preferably sight feed) must be fitted to each part requiring continuous lubrication.

6. If the engines are of the closed-in type, they must be so fitted that the contained lubricating oil can be drained when necessary, and in wood vessels an easily drained metal or metal-lined tray must be fitted to prevent leakage of either fuel oil or of lubricating oil from saturating the woodwork.

7. Carburettors, where petrol is used, and vaporisers, where paraffin is used, should be so designed that when the engine is stopped the fuel supply is automatically shut off. If an overflow is provided in the carburettor or vaporiser, a gauze-covered tray with means of draining it must be fitted to prevent the fuel from flowing into the bilges.

Strong metallic gauze diaphragms should be fitted either between the carburettor (or vaporiser) and cylinders, or at the air inlets.

8. If the ignition is electric, either by magneto or by coil and accumulator, all electric leads must be well insulated and suitably protected from mechanical injury. The leads should be kept remote from petrol pipes, and should not be placed where they may be brought into contact with oil.

The commutator must be enclosed; and the sparking coils must not be placed where they can be exposed to explosive vapours.

9. No exposed spark gap should be fitted.

10. In paraffin and heavy-oil engines where lamps are used for ignition or for vaporising, these lamps should be fixed by some suitable bracket, and the flame enclosed when in use.

11. The circulating pump sea suction is to have a cock or valve on the vessel's skin placed on the turn of the bilge in an easily accessible position, and the circulating pipe is to be provided with an efficient strainer inside the vessel. The discharge overboard is to be fitted with a cock or valve on the vessel's skin if it is situated under or near the load line of the vessel.

12. The pumping arrangements are to be the same as would be required in the case of a steam vessel of the same size and power, with the exception that no bilge injection need be fitted. In the cases of vessels fitted with water ballast, the water ballast pump must have one direct suction from the engine-room bilges in addition. In open launches, and in small sailing vessels in which the engines are auxiliary only, a suitable additional hand pump, fitted to draw from the engine-room bilges, may be accepted in lieu of a power-driven pump.

13. The cylinders are to be tested by hydraulic pressure to twice the working pressure to which they will be subjected. The water jackets of the cylinders to 50 lbs. per square inch, and the exhaust pipes and silencer to 10 lbs. per square inch.

Fuel Tanks and Connections.

Section 4. 1. Separate fuel tanks are to be tested with all fittings, to a head of at least 15 feet of water. If pressure-feed tanks are employed, they are to be tested to twice the working pressure which

will come on them, but at least to a head of 15 feet of water. If the tanks are made of iron or steel they should be galvanised.

2. Strong and readily removable metallic gauze diaphragms should be fitted at all openings on petrol tanks.

3. Paraffin or heavy-oil tanks, not used under pressure, are to be fitted with air pipes leading above deck. Pressure-feed tanks and tanks containing petrol should be provided with escape valves discharging into pipes leading to the atmosphere above deck. The upper ends of all air pipes are to be turned down, and pipes above 1 inch diameter are to be provided with gauze diaphragms at the end.

4. No glass gauges are to be fitted to fuel tanks containing either petrol, paraffin, or heavy oil.

5. Filling pipes are to be carried through the deck so that the gas displaced from the tanks has free escape to the atmosphere.

6. Separate fuel tanks should be provided with metal-lined trays to prevent any possible leakage from them flowing into the bilges or saturating woodwork. Arrangements are to be provided for emptying the tanks and draining the trays beneath them. For petrol tanks the trays must have drains leading overboard where possible, or they should be gauze-covered trays with means for draining them.

7. All fuel pipes are to be of annealed seamless copper with flexible bends. Their joints are to be conical, metal to metal. A cock or valve is to be fitted at each end of the pipe conveying the fuel from the tank to the carburettor or vaporiser. The fuel pipes should be led in positions where they are protected from mechanical injury and can be exposed to view throughout their whole length.

8. The engine-room, and the compartment in which the fuel tanks are situated are to be efficiently ventilated.

9. An approved fire-extinguishing apparatus must be supplied.

SUPERHEATED STEAM.

Previous to the general adoption of the compound engine on shipboard and the raising of the working pressure to 60 lbs. per square inch, superheaters were in very general use both in the Navy and Mercantile Marine. The old box form of boiler, made for a pressure of 30 lbs., had fitted in the uptake some form of tubular receiver for steam which was exposed to the heat action of the effluent gases, so that its temperature was raised from 274° F. to as much as 400°; generally, however, it was not possible to get so high as this, and in the Mercantile Marine 360° was satisfactory. The economy due to this conversion of waste heat into energy was marked, but the life of the superheater was never very long, and the restrictions imposed by the Board of Trade latterly were onerous; shipowners were therefore content with the saving effected by the compound system and the high-pressure boiler, and so gave up the use of what could have very materially improved the efficiency of the new installations. To-day the temperature of the saturated steam generated for quadruple engines is 390° F., or as high as superheated steam was formerly, and the absolute pressure is nearly five times; but, notwithstanding, even superheating it is an advantage and can be effected at comparatively small cost. In fact,

moderate superheat can be obtained still from the waste heat in the uptake; but to obtain the best economic results the grate area has to be such that the temperature in the uptake is considerably higher than is necessary for evaporative efficiency.

Table LXXIIIa.—A Comparison of the Steam Consumption with and without Superheat when Working with Steam at 180 Lbs. Pressure in Lbs. per S.H.P. Hour.

S.H.P.	Triple Compound Reciprocator.		Ordinary Direct Drive Turbine.		Combined Reciprocator and Turbine.	
	Vac. 26. Saturated Steam.	Vac. 26. Superheat to 572° F.	Vac. 28·5. Saturated Steam.	Vac. 28·5. Superheat to 572° F.	Vac. 28·5. Saturated Steam.	Vac. 28·5. Superheat to 572° F.
500	16·50	11·75	14·00	11·75	18 to 22	13 to 15
1000	15·25	11·25	11·75	9·75	12 to 13	9·5 to 9·9
1500	14·75	11·00	11·25	9·50	11·25	9·00
2000	14·50	11 00	11·00	9·25	11·00	8·70

The maximum temperature for safe and efficient working in every-day practice is 550°, or about 160° F. above that of the saturated steam. Some engineers on shore use steam at 600° or even a little higher, but in doing so there is risk which is hardly worth running. Moreover, to attain so high a superheat there must be an independently fired superheater, and the gases on leaving it must be very hot.

The specific heat of steam is determined by this formula.

$$\text{Specific heat at constant pressure} = 0{\cdot}4304 + 0{\cdot}000378 \times T,$$

wherein T is the temperature of the steam in degrees Fahr.

Table LXXIV.—Specific Heat of Steam at various Temperatures.

Temp.	Specific Heat.	Temp.	Specific Heat.	Temp.	Specific Heat.	Temp.	Specific Heat.	Temp.	Specific Heat.
F.°		F.°		F.°		F.°		F°.	
300	0·5438	380	0·5740	460	0·6043	540	0·6345	620	0·6648
310	0·5476	390	0·5778	470	0·6081	550	0·6383	630	0·6685
320	0·5514	400	0·5816	480	0·6118	560	0·6421	640	0·6723
330	0·5551	410	0·5854	490	0·6156	570	0·6459	650	0·6761
340	0·5589	420	0·5894	500	0·6194	580	0·6496	660	0·6799
350	0·5627	430	0·5929	510	0·6232	590	0·6534	670	0·6837
360	0·5665	440	0·5967	520	0·6270	600	0·6572	680	0·6874
370	0·5703	450	0·6005	530	0·6307	610	0·6610	690	0·6912

The amount of heat required to superheat a pound of steam to a temperature T_s from its natural temperature T may be ascertained by taking the specific heat at the mean temperature and multiplying by the difference in temperature. Thus:—

(*a*) $$\text{Mean temperature} = \frac{T + T_s}{2}.$$

(*b*) Amount of heat to raise temperature = sp. heat of $\frac{T + T_s}{2}$ $(T_s - T)$.

Example.—To superheat a pound of steam at a pressure of 180 lbs., or 195 absolute, to 500° F.

$$\text{Mean temperature} = \frac{380° + 500°}{2} = 440°.$$

Specific heat $440° = 0{\cdot}5967$.

Difference in temperature $= 500° - 390° = 110°$.

Heat required $= 0{\cdot}5967 \times 110°$ or $65{\cdot}637$ B.T.U.

The total heat of superheated steam may be calculated by Prof. Peabody's formula, which is as follows,—

$$\text{Total heat} = 0{\cdot}4805\ (T_a - 10{\cdot}38\sqrt{P}) + 857.$$

T_a is the absolute temperature Fahr.; P is the absolute pressure.

Or by Prof. Ripper's formula,

$$\text{Total heat} = H + 0{\cdot}48\ (t_s - t_1),$$

where H is the total heat of saturated steam at a pressure p and a temperature t_1, and t_s is the temperature to which it is raised by superheating.

Example.—The total heat of steam at 200 lbs. gauge pressure and 500° F. temperature will be

$$H = 0{\cdot}4805\left\{(500 + 461) - 10{\cdot}38\ (200 + 15)^{\frac{1}{2}}\right\} + 857.$$
$$= 442{\cdot}7 + 857 = 1299{\cdot}7° \text{ F.}$$

Superheated steam transmitted through pipes is greater than is the case with saturated steam of same pressure. The quantity may be estimated by the following formula.

d is the diameter o the pipe in inches, whose length is L feet; W is the weight of a cubic foot of the steam at a pressure P_1 at entry and P_2 at exit

$$\text{Weight passed per minute} = 87\sqrt{\frac{W(P_1 - P_2)d^5}{L\left(1 + \frac{3{\cdot}6}{d}\right)}}.$$

Table LXXV.—Maximum Work possible from 1 lb. of Superheated Steam expanding to and exhausting at 1 lb. pressure B.T.U.

Initial Pressure Absolute.	Temperature of Steam after superheat, degrees Fahr.													
	350	375	400	425	450	475	500	525	550	575	600	625	650	675
Lbs.														
100	300·0	315·9	329·0	344·0	359·0	374·0	389·5	405·0	421·0	437·0	453·0	470·0	486·5	503·0
125	306·5	321·5	335·0	349·5	364·0	379·0	394·5	410·0	426·0	442·0	458·0	475·0	491·3	508·0
150	..	326·0	339·0	353·0	368·0	383·0	398·0	414·0	430·0	446·0	462·0	478·5	495·0	512·0
175	..	329·3	342·0	356·5	375·0	386·5	401·0	417·3	433·0	449·3	465·3	482·0	498·5	515·5
200	..	332·0	344·5	358·5	373·5	389·0	404·0	420·0	436·0	452·0	468·0	484·0	501·0	518·0
225	..	..	346·4	360·5	355·5	391·0	406·0	422·0	438·0	454·0	470·0	486·3	502·7	520·0
250	..	..	348·0	362·0	377·5	392·5	408·9	423·5	439·5	455·0	471·5	488·0	504·0	521·0
275	..	..	..	363·0	379·0	394·0	409·5	425·0	441·0	456·4	472·8	489·3	505·0	522·0
300	..	..	..	364·0	380·0	395·0	410·5	426·0	442·0	457·1	473·4	490·0	506·6	522·6

Table LXXVa.—Weight of Superheated Steam passed per minute in lbs. through Pipes 240 diameters long, with a drop of 1 lb. Pressure.

Gauge Pressure.	Diameter of Pipe in Inches.									
	1	2	3	4	5	6	7	8	9	10
Lbs.										
100	5·25	26·0	64·2	118·5	196	293	400	535	690	863
120	5·63	27·9	69·9	127·2	210	315	433	574	740	925
140	6·00	29·7	73·2	135·0	223	334	458	609	786	980
160	6·31	31·3	77·4	142·0	235	352	483	640	825	1033
180	6·64	33·0	81·5	148·9	245	368	503	670	865	1080
200	6·95	34·5	85·7	155·8	256	383	525	698	903	1125
220	7·25	36·0	89·8	162·3	266	398	545	726	940	1172
240	7·55	37·5	93·8	168·4	275	412	563	752	983	1214

The heating surface of a superheater per pound of steam must depend on the difference in temperature between the hot gases and the steam.

With considerable difference 6 B.T.U.'s are transferred per square foot of surface per hour for each degree of difference Fahr., when the surfaces within and without are clean.

That is, if steam of 200 lbs. pressure, 500° temperature, is passing through a superheater exposed to hot gas at 650° temperature,

Each square foot should pass

$$6\ (650 - 500) \text{ or } 900 \text{ B.T.U. per hour.}$$

Each pound of steam requires (500 − 387) 0·599 = 67·7 B.T.U., so that if the consumption of steam is 12 lbs. per I.H.P., the surface per I.H.P. = 67·7 × 12 ÷ 900, or 0·9027 square feet.

SKIN FITTINGS AND VALVES.

Blow-off valves.—These should always be bronze stop-valves opening inwards, and should have spigots passing through the ship's plating. If any guard cock is fitted, it should be placed between the valve on the boiler and the sea-valve, and independent of both.

The method of attachment to the ship's skin, recommended by Lloyd's, is shown in figs. 43, and 44; the Admiralty method in figs. 45 and 46.

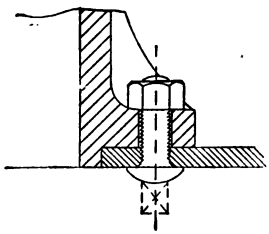


FIG. 43.

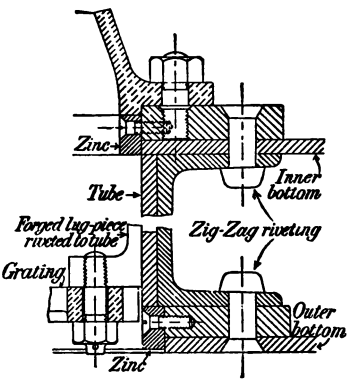


FIG. 44.

Where the valve has to open a communication through a double bottom, as is usually the case in Naval work, one of the Kingston type is generally employed, attached to the outer bottom, as shown in fig. 46, and made tight by means of a stuffing box, where it passes through the inner bottom.

The use of cast-iron for boxes or casings of blow-off valves is forbidden by the Board of Trade rules.

Discharge valves.—Here also, with the object of reducing the number of openings in the ship's skin, the smaller valves should, where possible, discharge into the casings of the larger ones, the outlets from the latter being increased in area as necessary.

In the Merchant Service, lifting non-return valves with cast-iron casings are generally used; but in the Navy, though the same type of valve is used, the only material permitted is bronze.

Provision should be made for hanging up the valve, but, when the circulating pumps are of the reciprocating type, it should not be possible to hold it on its seat in any way.

All discharge valves should have spindles passing through the covers, and cross handles for turning them round on their seats.

In some cases, where room is pinched, a swinging flap-valve may be employed with advantage; and where the pumps are of the centrifugal type, either valves of this class or sluice valves should be used, in order to reduce the resistances as much as possible.

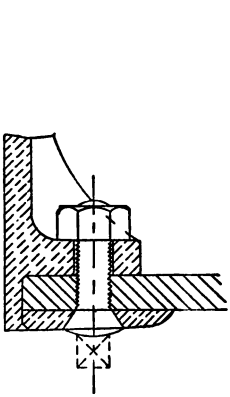


FIG. 45.

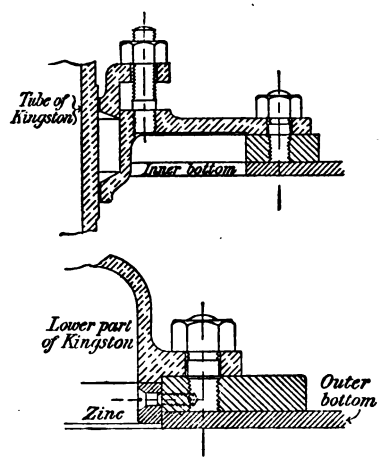


FIG. 46.

There should always be a spigot passing through the skin plating of the vessel, and, where the valve is large, it is often necessary to have a compensating plate inside riveted to the skin plate.

The Board of Trade regulations referring to sea connections are as follows:—

194. All inlets or outlets in the bottom or side of a vessel near to, at, or below the deep-load water line other than the outlets of water-closet, soil, scupper, lavatory, and urinal pipes should have cocks or valves fitted between the pipes and the ship's side or bottom; such cocks or valves should be attached to the skin of the ship, and be so arranged that they can be easily and expeditiously opened or closed at any time; and the cocks, valves, and the whole length of the pipes should be accessible at all times.

Cocks or valves standing exceptional distances from the ship's plating, that is where the necks are longer than is necessary for making the joint, should not be passed without the sanction of the Board of Trade, and one condition of their being passed is that they should be made of gun-metal, and well bracketed.

188. With a view to the prevention of accidents to boilers through the blow-off cocks being left open after the boiler is run up, and to prevent water getting accidentally into the ship by cocks being left open, all blow-off cocks and sea connections below the plates or out of sight should be fitted with a guard over the plug, with a featherway in the same, and a key on the spanner, so that the spanner cannot be removed unless the cock is closed. The spanner, &c., when in place should extend to the platform. When cocks are in sight, guards need not be fitted provided the spanners are secured to the plugs by pins. The spanners should not be shrunk on the heads of the plugs. One cock should be fitted to the boiler, and another cock on the skin of the ship or on the side of the Kingston valve. Valves may be substituted for the blow-off cocks on the boilers, but, in such cases, the blow-off cock on the skin of the ship must be fitted with a spanner guard so that the spanner cannot be removed when the cock is open; and, if the blow-off pipe is used for more than one boiler, an intermediate switch-cock, or suitable non-return valve to each boiler, should be fitted so that water cannot be blown from one boiler to another.

189. When the pipes are so arranged that the water in the boilers can be circulated by means of the donkey pump, similar precautions should be observed; and a cock fitted with spanner and guard, the handle of which will stand above the level of the platform, should be fitted in the circulating pipes, preferably near the pump.

190. A non-return valve having a screw spindle not attached, by which the valve may be set down on its seat when necessary, should be fitted to all pipes which are so led or placed that water could, unless such non-return valves were fitted, run from the boiler or the sea into the bilge, either by accidentally or intentionally leaving a cock or valve open; the only exception to this requirement has reference to the fireman's ash cock, which must have a cock or valve on the ship's side and be above the stoke-hold plates.

191. Cast-iron stand-pipes or cocks intended for the passage through them of hot brine should not be passed.

Surveyors should also discourage the use of cast-iron chocks and saddles for boilers, and particular attention should be paid to the chocking of boilers, more especially when they are fired athwartships.

192. The exhaust pipe for the donkey engine should not be led through the ship's side, but should be led on deck or into the main waste steam pipe,* and in all cases should have a drain-cock on it.

193. In the case of the outlets of water-closet, soil, scupper, lavatory and urinal pipes which are below the weather deck, there should be an

* AUTHORS' NOTE.—This plan renders overhaul of main safety-valves difficult when vessel is in port; a separate waste pipe should always be fitted.

elbow of good substantial metal other than cast-iron or lead ; and the pipe connected with this elbow should, if of lead, have a sufficient bend to provide for expansion in the pipe or any movement from the working of the ship. Pipes, no matter of what material they may be constructed, should never be fitted in a direct line between the apertures in the ship's side and their connection with the deck, or closet, or other fittings. The pipes and valves should be protected from the cargo by a substantial casing of wood or iron, which need not be watertight.

Lloyd's Rules relating to sea connections are as follows :—

5. Cocks and valves connecting all suction pipes to be fixed above the stokehold and engine-room platforms.

6. The arrangements of pumps, bilge injections, suction and delivery pipes, is to be such as will not permit of water being run from the sea into the vessel by an act of carelessness or neglect.

Sec. 7.—8. All discharge pipes to be, if possible, carried above the deep-load line, and to have discharge valves fitted on the plating of the vessel in an accessible position.

Sec. 7.—9. No pipes to be carried through the bunkers without being properly protected.

Sec. 9.—2. All sea-cocks to be fitted on the plating of the vessel above the level of the stoke-hole and engine-room platforms, or attached to Kingston valves of a height sufficient to lift them up to the level of these platforms.

36. The bolts securing all cocks or sea connections to the plating of the vessel are to be tapped into the plating of the vessel or fitted with countersunk heads.

37. The blow-off cocks on the plating of the vessel are to be fitted with spigots passing through the plating, and a brass or gun-metal ring on the outside. The cocks are to be so constructed that the key or spanner can only be taken off when the cock is shut.

See also paragraph 6 of section 27 of Lloyd's rules.

Table LXXVI.—Results of Trials of Triple-Expansion Three-Crank Engines.

Ship.	Number of Screws.	Cylinders.				Engine Room.						One Engine Only.			Nominal Rate of Expansion.
		Diameters.			Stroke.	Steam.	Vacuum.	Revolutions.	Cut-off in H.P. Cylinders.	Mean Pressures.			Indicated Horse-power.			
		H.P.	M.P.	L.P.						H.P.	M.P.	L.P.	H.P.	M.P.	L.P.	Total.	
		ins.	ins.	ins.	ins.	lbs.	ins.			lbs.	lbs.	lbs.					
S.S. Ag. V.	Twin	41	67	106	63	149	24·0	78·0	0·78	62·6	27·8	11·16	2214	2470	2494	7178	8·53
S.S. Krsn.	,,	36	57	92	54	155	27·3	90·6	..	82·9	36·8	14·2	2085	2320	2335	6740	..
S.S. Lgra.	,,	33	54	88	60	183	25	72·0	..	77·5	30·0	12·0	1444	1490	1692	4535	..
H.M.S. Ldn.	,,	31¼	51¾	84	51	245	25·5	110	..	115·0	44·0	17·9	2543	2601	2815	7859	..
H.M.S. Argf.	,,	26	42	68	39	245	26·0	143	0·70	112·5	47·1	20·5	1685	1841	2100	5406	9·80
S.S. Czgo.	,,	32	48	80	48	145	26·0	71·0	0·66	68·1	32·0	11·0	943	994	947	2884	9·50
S.S. Otlo.	,,	24¾	40	70	43	200	28·0	71·0	0·59	88·5	36·8	13·6	733	794	903	2430	15·25
S.S. R.W.	Single	25	40	68	45	180	27·5	61·0	..	68·7	27·1	10·15	500	502	545	1547	7·80
S.S. Vna.	Twin	26	39½	61	36	155	26·5	135	0·72	62·4	31·2	14·15	810	933	1011	2754	7·80
S.S. Ped.	,,	19	30	48	24	180	25·0	192	0·68	81·0	33·5	13·7	536	564	577	1067	9·37
S.S. Ostr.	Single	19	33	56	36	200	25·0	91·5	0·59	87·8	33·5	10·9	325	417	470	445	1333	14·70
H.M.S. Snpr.	Twin	19¼	28½	43	18	180	25·0	371	0·58	64·5	28·8	14·6	395	635	592	715	1942	8·62
H.M.S. Chgr.	,,	18	26	39½	18	160	25·0	363	0·81	62·0	37·8	13·5	41·9	523	626	547	1696	6·00
S.S. Mipo.	Single	13	22	36	24	200	23·5	112	0·60	96·7	38·7	13·12	40·0	174	199	181	554	12·75
S.S. Iona.	,,	22	34	57	39	150	27·8	61·5	0·25	47·0	20·0	7·2	21·7	208	217	220	645	26·88
S.S. Arte.	,,	12	18	30	18	155	28·0	136	0·62	67·8	29·1	11·4	32·5	92·1	91·4	99·5	283	10·1
S.S. Zde.	Twin	12¼	19	32	24	175	27·0	148	0·63	69·0	36·5	13·75	36·8	152	182	197	531	9·62
S.S. Gte.	Single	12	20	32	23	200	22·0	136	0·58	74·5	38·8	15·50	41·0	133	192	197	522	12·26

Table LXXVI. (continued).—Results of Trials of Triple-Expansion Four-Crank Engines.

Ship.	Number of Screws.	Cylinders. Diameter. H.P.	M.P.	L.P.	L.P.	Stroke.	Engine Room. Steam.	Vacuum.	Revolutions.	Cut-off in H.P Cylinders.	Mean Pressures. H.P.	M.P.	L.P.	L.P.	Ref. L.P.	One Engine only. Indicated Horse-power. H.P.	M.P.	L.P.	L.P.	Total.	Nominal Rate of Expansion.
		ins.	ins.	ins.	ins.	ins.	lbs.	ins.			lbs.	lbs.	lbs.	lbs.	lbs.						
H.M.S. Gh. .	Twin	43½	71	81½	81½	48	278	27·0	126	..	105	44·6	16·9	17·9	50·8	4764	5399	2698	2849	15,710	..
H.M.S. Wr.	,,	43½	69	77	77	42	194	26·7	138	..	72·6	32·7	18·5	19·2	44·1	3147	3568	2513	2607	11,834	..
H.M.S. Am.	,,	41½	65½	73½	73½	42	185	25·0	143	..	82·6	32·5	16·7	16·6	..	3383	3315	2143	2136	10,977	..
H.M.S. Cd. .	,,	37	60	69	69	42	296	26·5	145	..	105·2	41·3	19·1	18·8	49·3	3484	3594	2194	2170	11,442	..
H.M.S. Dm.	,,	34	55½	64	64	48	245	27·0	120	..	90·0	34·0	21·9	21·7	50·3	2370	2386	2044	2025	8,825	..
H.M.S. Fd. .	,,	31½	50½	57	57	30	227	25·3	212	..	109	34·5	14·9	14·8	..	2727	2224	1218	1217	7,386	..
H.M.S. Tz. .	,,	24½	38½	42½	42½	24	271	24·0	246	..	111	47·2	21·5	22·7	57·3	1534	1637	897	946	5,014	..
S.S. Par. .	,,	26	40	45	45	30	177	27·0	177	0·69	57·5	30·3	15·4	15·3	36·9	818	1019	682	630	3,149	8·89
S.S. Cr. .	,,	14½	23	27	27	33	180	26·5	145	0·75	83	38·0	18·4	18·2	44·6	331	382	255	252	1,220	9·33
H.M.S. Dv. .	,,	19½	30	34	34	18	206	25·0	385	..	77·3	45·5	17·7	17·2	..	808	1127	563	547	3,045	..
H.M.S. Sr. .	,,	18¾	27½	28	28	18	194	24·1	391	..	63·7	33·0	19·0	18·7	..	626	684	412	406	2,128	..

Table LXXVI.—Results of Trials of Quadruple Expansion Engines.

Ship.	Number of Screws.	Cylinders.					Engine Room Records.										One Engine only.					Ratio of L.P. to H.P. Cylinders.	Nominal Rate of Expansion.
		Diameters.				Piston Stroke.	Steam Pressure.	Vacuum.	Revolutions.	Cut-off in H.P. Cylinders.	Mean Pressures.						Indicated Horse-power.						
		H.P.	M.P$_1$	M.P$_2$	L.P.						H.P.	M.P$_1$	M.P$_2$	L.P.	Refd to L.P.		H.P.	M.P$_1$	M.P$_2$	L.P.	Total.		
		ins.	ins.	ins.	ins.	ins.	lbs.	ins.			lbs.	lbs.	lbs.	lbs.	lbs.								
J.B.C 1.	2	39	54·5	77	110	66	195	26·9	89	0·77	70	39·6	18·5	11·8	39·1		2495	2755	2575	3330	11,155	8·0	10·4
N.E.M. 1	1	28·5	41	58	84	54	220	25·3	70	0·67	85	40	22·3	10·0	39·9		1030	1010	1122	1058	4,220	8·69	13·0
J.B.C. 2	2	28·7	41	58·5	84	60	197	27·0	81	0·78	85·2	40	19·4	10·2	37·7		1357	1279	1280	1392	5,308	8·53	11·0
D.R. 1	1	28	40	57	82	57	220	27·0	74	0·83	83·4	35·2	19·0	9·66	37·0		1079	942	1035	1087	4,143	8·58	10·39
J.B.C. 3	1	26	39	56	78·5	54	210	26·1	83·7	0·66	73·8	30·3	15·4	9·4	34·2		888	982	869	1037	3,776	9·11	13·8
D.R. 2	1	26	38	55	78	54	220	27·0	76	0·68	79·5	30·0	18·3	11·5	36·6		876	705	903	1139	3,623	9·00	13·23
D.R. 3	2	24·5	35	50	72	51	220	26·5	78·5	0·76	84·8	37·6	23·2	12·0	41·9		807	731	923	989	3,450	8·64	11·4
E.S.B.	1	20	28	39·5	57	33	212	25·0	140	0·63	73·5	37·0	19·8	10·0	40·0		540	532	565	596	2,233	8·12	12·9
N.E.M. 2	1	23	32·5	47	68	48	220	25·5	71	0·65	78·7	47·1	15·7	12·9	40·2		556	676	470	802	2,504	8·74	13·4
H.L.	1	22·5	31·5	45	66	51	215	25·5	76	0·57	74	35·5	18·3	8·3	33·3		570	542	566	550	2,228	8·70	15·1
D.R. 4	2	19·5	27·5	40	58	48	220	26·5	84	0·81	87·4	48·4	24·5	14·0	46·3		531	584	625	752	2,492	8·85	10·96
E.S.B.	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..		..	..	..	..	..	..	..

Rules for Copper and Steel Pipes adopted by the Board of Trade and the Register Societies:—

(1) Steam and other pipes may be made of wrought iron or of wrought steel made in an open-hearth furnace.

(2) When they are welded the process shall be by hammering or rolling the joint.

(3) On completion of any work which involves heating whether for welding the joint, welding on flanges, hot bending the pipe, or for any other purpose, the pipe shall be carefully annealed.

(4) Mild steel for lap-welded steam pipes may have a tensile strength not exceeding 28 tons per square inch, with a minimum elongation of 25 per cent. in 8 inches.

(5) Feed pipes if made of steel should be solid cold-drawn finished.

(6) All iron or steel pipes prior to being fitted in place shall be subjected to a hydraulic test of at least three times the working pressure, and all feed delivery pipes of iron or steel to four times.

(7) The working pressure (W.P.) allowed on pipes shall be determined by the following rules, where

D is the internal diameter in inches,
t is the thickness in 100ths of an inch.

(*a*) Solid cold-drawn cold finished pipes:—$\text{W.P.} = \frac{(t-10)}{\text{D}} \times 120.$

(*b*) Solid hot-drawn pipes:—$\text{W.P.} = \frac{(t-12)}{\text{D}} \times 120.$

(*c*) Welded pipes of iron or steel:—$\text{W.P.} = \frac{(t-12)}{\text{D}} \times 90.$

(*d*) Feed pipes (delivery):—Boiler pressure $= \frac{(t-10)}{\text{D}} \times 100.$

(8) In all steam pipes provision is to be made for expansion and contraction to take place without unduly straining them.

Copper Pipes should contain not less than 99·25 per cent. of copper and from 0·25 to 0·45 of arsenic, and must not be made from the electro deposition.

Copper, steam, feed, blow-off, and scum pipes must be properly annealed before putting in place, and all subject to a working pressure over 75 lbs. per sq. in. shall be solid-drawn.

Copper steam pipes on completion shall be subjected to hydraulic test to at least twice the W.P.

For all *Feed Pipes* it shall be 2·5 × W.P.

Strips cut from copper tubes should have an ultimate tensile strength of at least 14 tons per square inch with an elongation of 40 per cent. in 2 inches after annealing and quenching in water.

$$\text{The working pressure} = \frac{(t-3)}{\text{D}} \times \text{F}.$$

t is the thickness in 100ths of an inch. D is the bore in inches. F for solid-drawn pipes is 60, and for brazed 45.

Copper pipes, when bent, must be thicker than given above; the radius of curvature at centre line must not be less than twice the outside diameter.

Table LXXVII.—Wire Gauges and their Equivalents.

L. S. G. Legal Standard Wire Gauge.		B. W. G. Birmingham Wire Gauge.		W. W. G. Whitworth Wire Gauge.	
No. of Gauge.	Equivalent in ins.	No of Gauge.	Equivalent in ins.	No. of Gauge.	Equivalent in ins.
7/0	·500	5/0	·500	300	·300
6/0	·464	4/0	·460	280	·280
5/0	·432	3/0	·420	260	·260
4/0	·400	2/0	·380	250	·250
3/0	·372	1/0	·340	240	·240
2/0	·348	1	·300	220	·220
0	·324	2	·280	200	·200
1	·300	3	·260	180	·180
2	·276	4	·240	165	·165
3	·252	5	·220	150	·150
4	·232	6	·200	135	·135
5	·212	7	·180	125	·125
6	·192	8	·164	120	·120
7	·176	9	·148	110	·110
8	·160	10	·132	100	·100
9	·144	11	·120	95	·095
10	·128	12	·108	90	·090
11	·116	13	·096	85	·085
12	·104	14	·084	80	·080
13	·092	15	·072	75	·075
14	·080	16	·064	70	·070
15	·072	17	·056	65	·065
16	·064	18	·048	60	·060
17	·056	19	040	55	·055
18	·048	20	·036	50	·050
19	·040	21	·032	45	·045
20	·036	22	·028	40	·040
21	·032	23	·024	38	·038
22	·028	24	·022	36	·036
23	·024	25	·020	34	·034
24	·022	26	·018	32	·032
25	·020	27	·016	30	·030
26	·018	28	·014	28	·028
27	·0164	29	·013	26	·026
28	·0148	30	·012	24	·024
29	·0136	31	·011	22	·022
30	·0124	32	·010	20	·020
31	·0116	33	·009	19	·019
32	·0108	34	·008	18	·018
33	·0100	35	·007	17	·017
34	·0092	36	·006	16	·016
35	·0084	37	·005	15	·015
36	·0076	38	·004	14	·014
37	·0068	39	·003	13	·013
38	·0060	40	·002	12	·012

The following Table shows the Legal Standard Wire Gauge and its equivalents in millimetres:—

Table LXXVIII.—Legal Standard Wire Gauge and Metric Equivalents.

Number of Gauge.	Equivalents.		Number of Gauge.	Equivalents.	
	Inch.	Millimetre.		Inch.	Millimetre.
7/0	·500	12·700	23	·024	·610
6/0	·464	11·785	24	·022	·559
5/0	·432	10·973	25	·020	·508
4/0	·400	10·160	26	·018	·457
3/0	·372	9·449	27	·0164	·4166
2/0	·348	8·839	28	·0148	·3759
0	·324	8·229	29	·0136	·3454
1	·300	7·620	30	·0124	·3150
2	·276	7·010	31	·0116	·2946
3	·252	6·401	32	·0108	·2743
4	·232	5·893	33	·0100	·2540
5	·212	5·385	34	·0092	·2337
6	·192	4·877	35	·0084	·2134
7	·176	4·470	36	·0076	·1930
8	·160	4·064	37	·0068	·1727
9	·144	3·658	38	·0060	·1524
10	·128	3·251	39	·0052	·1321
11	·116	2·946	40	·0048	·1219
12	·104	2·642	41	·0044	·1118
13	·092	2·337	42	·0040	·1016
14	·080	2·032	43	·0036	·0914
15	·072	1·829	44	·0032	·0813
16	·064	1·626	45	·0028	·0711
17	·056	1·422	46	·0024	·0610
18	·048	1·219	47	·0020	·0508
19	·040	1·016	48	·0016	·0406
20	·036	·914	49	·0012	·0305
21	·032	·813	50	·0010	·0254
22	·028	·711	...	...	...

Table LXXIX.—Thicknesses of Copper Pipes (L.S.G.).

(May be taken as giving thoroughly reliable strengths for all ordinary marine work.)

Diameter of pipe in inches.	Steam pipes. Boiler pressures in lbs.						Auxiliary exhaust pipes.	Waste steam pipes.	Main water pipes.	Bilge suction and diach. feed suction and fire service.	Diameter of pipe in inches.	Main eduction and air-pump suction.
	200	180	155	125	85	50						
22	...	...	...	...	0	7	...	...	4	...	35	3
21	...	...	...	...	1	7	...	...	4	...	33	3
20	...	...	...	...	1	7	...	...	5	...	32	4
19	...	...	...	...	2	8	..	...	5	...	29	4
18	...	...	...	4/0	2	8	.	13	5	...	28	5
17	...	...	6/0	3/0	3	8	...	13	6	...	25	5
16	...	7/0	5/0	2/0	3	8	...	14	6	...	24	6
15	...	6/0	4/0	0	4	8	...	14	7	...	21	6
14	7/0	5/0	3/0	0	4	9	11	14	7	...	20	7
13	6/0	4/0	2/0	1	5	9	11	14	7	...	17	7
12	5/0	3/0	0	2	5	9	11	14	8	...	16	8
11	4/0	2/0	1	3	6	9	11	15	8	...	15	8
10	3/0	0	1	3	6	10	11	15	9	...	14	9
9	2/0	1	2	4	7	10	11	15	9	...	13	9
8	1	2	3	5	8	10	12	15	9	...	12	10
7	2	3	4	6	8	11	12	15	10	9	11	10
6	3	4	5	7	9	11	12	15	10	10	10	11
5	5	6	6	8	9	11	12	15	11	10	9	11
4	6	7	7	9	10	12	12	15	11	11	8	12
3	8	9	9	10	11	12	12	15	12	11	7	12
2	10	11	11	11	12	12	12	15	12	12	6	13
1	13	13	13	13	13	13	13	15	13	13	5	13

	Blow-off and scum pipes.								
Diameter of pipe in inches, . .	1	1¼	1½	1¾	2	2¼	2½	2¾	3
Thickness, L.S.G.	10	10	10	9	9	8	8	7	7

Feed discharge pipes to be as steam pipes for 30 per cent. higher pressure; but in no case to be taken lower than 125 lbs.

Receiver pipes to be as steam pipes for half the test pressure of the cylinder to which they lead steam; but in no case to be taken lower than 50 lbs.

The above gauges refer to straight pipes only; bends to be suitably strengthened.

Table LXXX.—Solid-drawn Copper Pipes: Working Pressures allowed in them by the Rule of B.M.E.D. & C. Committee.

Diameter of Bore.	Thickness in Legal Standard Wire Gauge.													
	18.	17.	16.	15.	14.	13.	12.	11.	10.	9.	8.	7.	6.	5.
ins.														
1·0	108	156	204	252	300	...	...	...	...	...	...	...	...	...
1·25	86	124	163	201	240	297	...	...	...	...	...	...	...	...
1·5	72	104	136	168	200	248	296	...	...	...	...	...	...	...
1·75	62	89	116	144	172	212	253	295	...	...	...	...	...	...
2·0	...	78	102	126	150	186	222	258	294	...	...	...	...	...
2·25	...	69	91	112	133	165	198	228	260	304	...	...	...	...
2·5	...	62	81	101	120	152	177	206	235	273	312	...	...	...
2·75	...	...	74	91	109	135	161	187	213	248	283	319	...	...
3·0	...	...	68	84	100	124	148	172	196	228	260	292	324	...
3·25	...	...	62	77	92	114	136	158	180	209	239	269	299	...
3·5	...	...	...	72	85	106	127	147	168	195	223	250	277	312
3·75	...	...	...	67	80	99	119	138	156	180	208	233	259	291
4·0	...	...	...	63	75	93	111	129	147	171	195	219	243	273
4·25	...	...	...	...	70	85	104	121	138	164	184	206	228	257
4·5	...	...	...	...	67	80	98	114	130	152	173	194	216	242
5·0	...	...	...	...	60	75	89	103	117	137	156	175	194	218
5·5	...	...	...	...	...	68	81	94	107	124	142	159	176	198
6·0	...	...	...	...	...	62	74	86	96	116	130	146	162	182
6·5	...	...	...	...	...	...	68	80	89	107	120	134	149	168
7·0	...	...	...	...	...	...	63	74	82	100	112	125	139	156
7·5	...	...	...	...	...	...	...	69	77	93	104	117	129	145
8·0	...	...	...	...	...	...	...	64	72	87	98	109	121	136

N.B.—For the working pressure of brazed pipes multiply by 0·75 or divide by 1·33.

F the diameter of flange for a pipe whose bore is D and has bolts *d* in diameter.

$F = D + 5d$.

Diameter of bolt circle $D + 4d$.

Thickness of flange in 32nds of an inch $= 0{\cdot}55\sqrt{F \times W.\ Press.} + 6$.

The thicknesses of wrought-iron lap-welded pipes requisite to comply with the B.M.E.D. & C. Committee rule are shown in the following Table:—

Table LXXXI.—Welded Iron and Steel Pipes (v. page 281): Working Pressure allowed in them by the Rule of the Board of Trade, Lloyd's, etc.

Diameter of Bore.	Thickness in 100ths of an Inch.													
	20.	22.	24.	26.	28.	30.	32.	34.	36.	38.	40.	42.	44.	46.
ins.														
5·0	144	180	216	252	288	324	...	...	...	...	...	...	...	...
5·5	130	173	206	239	271	304	337	...	...	...	...	...	...	...
6·0	120	150	180	210	240	270	300	330	...	...	...	...	...	...
6·5	110	138	165	193	220	248	276	303	331	...	...	...	...	...
7·0	103	128	154	180	205	231	257	283	308	334	...	...	...	...
7·5	96	120	144	168	192	216	240	264	288	312	336	...	...	...
8·0	90	112	135	157	180	202	225	247	270	292	315	337	...	...
8·5	84	106	127	148	169	190	211	233	254	275	296	317	339	...
9·0	80	100	120	140	160	180	200	220	240	260	280	300	320	340
9·5	75	94	113	132	151	170	189	208	227	246	265	284	300	319
10	...	90	108	126	144	162	180	198	216	234	252	270	288	306
11	...	82	98	114	130	147	163	180	196	212	229	245	262	278
12	...	75	90	105	120	135	150	165	180	195	210	225	240	255
13	...	...	83	96	110	124	138	152	166	180	194	208	220	235
14	...	...	77	90	102	115	128	141	154	167	180	193	205	218
15	..	...	72	84	96	108	120	132	144	156	168	180	192	204
16	...	...	...	78	90	101	112	123	134	146	157	169	180	190
17	...	...	...	74	85	95	106	116	127	138	148	159	169	180
18	...	...	...	...	80	90	100	110	120	130	140	150	160	170
19	...	...	...	...	76	85	94	104	113	123	133	142	150	160
20	...	...	...	...	...	81	90	99	108	117	126	135	143	153
21	...	...	...	...	...	77	85	94	103	111	120	129	136	145
22	...	...	...	...	...	...	82	90	98	106	115	123	131	139
23	...	...	...	...	...	...	78	86	94	101	110	118	126	133
24	...	...	...	...	...	...	75	83	91	97	105	113	120	128

F the diameter of flanges for these pipes whose bore is D and the diameter of the jointing bolts is d.

$F = D + 6d$.

Diameter of bolt circle $= D + 4d$.

Thickness of flange in 32nds of an inch $= 0{\cdot}55\sqrt{F \times W.\ Press.} + 8$.

Table LXXXII.—Flanges for Copper Pipes.

Diameter of pipe in inches.	Diameter of flange.	Thickness of flange.		Diameter of bolts.	Radius of bolt circle.	Number of bolts.			Pitch of bolts. (nearest sixteenth.)		
		Steam, feed, blow-off, &c.	Auxiliary exhaust, water pipes, waste and eduction, &c.			Steam, feed, blow-off, &c.	Auxiliary exhaust and water pipes.	Waste and eduction pipes.	Steam, feed, blow-off, &c.	Auxiliary exhaust and water pipes.	Waste and eduction pipes.
ins.	ins.	in.	in.	in.	ins.				ins.	ins.	ins.
1	4	1/2	3/8	1/2	1 3/8	4	4	3	1 15/16	1 15/16	2 3/8
1 1/4	4 1/4	"	"	"	1 1/2	4	4	3	2 1/8	2 1/8	2 5/8
1 1/2	4 1/2	9/16	7/16	"	1 5/8	4	4	4	2 1/16	2 5/16	2 7/16
1 3/4	4 3/4	"	"	"	1 3/4	5	4	4	2 1/16	2 1/2	2 1/2
2	5	"	"	"	1 7/8	5	5	4	2 3/16	2 3/16	2 5/8
2 1/4	5 7/8	5/8	1/2	5/8	2 1/4	5	5	4	2 5/8	2 5/8	3 3/16
2 1/2	6 1/8	"	"	"	2 3/8	6	5	4	2 7/8	2 13/16	3 3/8
2 3/4	6 3/8	"	"	"	2 1/2	6	5	4	2 1/2	2 15/16	3 1/2
3	6 5/8	11/16	9/16	"	2 5/8	6	5	5	2 5/8	3 1/16	3 1/16
3 1/4	6 7/8	"	"	"	2 3/4	7	6	5	2 3/8	2 3/4	3 1/4
3 1/2	7 1/8	"	"	"	2 7/8	7	6	5	2 1/2	2 7/8	3 3/8
3 3/4	7 3/8	"	"	"	3	7	6	5	2 5/8	3	3 1/2
4	7 5/8	3/4	5/8	"	3 1/8	8	6	6	2 7/8	3 1/8	3 7/8
4 1/2	8 1/8	"	"	"	3 3/8	8	7	6	2 9/16	2 15/16	3 3/8
5	8 5/8	"	"	"	3 5/8	9	7	7	2 1/2	3 1/8	3 7/8
5 1/2	9 1/8	"	"	"	3 7/8	9	8	7	2 5/8	2 15/16	3 3/8
6	10 3/8	13/16	11/16	3/4	4 3/8	9	7	6	3	3 1 3/16	4 3/8
6 1/2	10 7/8	"	"	"	4 5/8	9	7	6	3 5/16	4	4 5/8
7	11 3/8	"	"	"	4 7/8	10	7	6	3	4 3/16	4 7/8
7 1/2	11 7/8	"	"	"	5 1/8	10	8	7	3 3/16	3 15/16	4 7/16
8	12 3/8	7/8	3/4	"	5 3/8	11	8	7	3 1/16	4 1/8	4 5/8
8 1/2	12 7/8	"	"	"	5 5/8	11	9	7	3 5/16	3 7/8	4 7/8
9	13 3/8	"	"	"	5 7/8	12	9	8	3 1/16	4	4 1/2
9 1/2	13 7/8	"	"	"	6 1/8	12	9	8	3 5/16	4 3/16	4 11/16
10	14 3/8	"	"	"	6 3/8	13	10	9	3 1/16	3 15/16	4 3/8
11	16 1/4	15/16	15/16	7/8	7 3/16	13	10	9	3 7/16	4 1/2	5
12	17 1/4	"	"	"	7 11/16	14	11	10	3 7/16	4 3/8	4 7/8
13	18 1/4	"	"	"	8 3/16	16	12	10	3 5/16	4 5/16	5 1/8
14	19 1/4	"	"	"	8 11/16	16	13	11	3 7/16	4 3/16	5
15	20 1/4	"	"	"	9 3/16	17	14	12	3 3/8	4 1/8	4 7/8
16	21 1/2	1	7/8	"	9 13/16	18	14	12	3 7/16	4 3/8	5 1/8
17	22 1/2	"	"	"	10 5/16	19	14	12	3 7/16	4 5/8	5 3/8
18	23 1/2	"	"	"	10 13/16	20	15	13	3 3/8	4 1/2	5 5/16

NOTE.—For pressures between 10 and 300 lbs., pitch of bolts is given by—Pitch = $\frac{10\sqrt[3]{dt^2}}{\sqrt[4]{P}} + \cdot 5$, where t = thickness of flange, d = dia. of bolt, and P = pressure

The flanges, of which particulars are given in the above Table, are designed either for coupling copper pipe to copper pipe, or copper pipe to a bronze casting of corresponding strength, as indicated in Fig. 47; for connections with cast-iron pipes or valves the flange must be larger, unless studs are used.

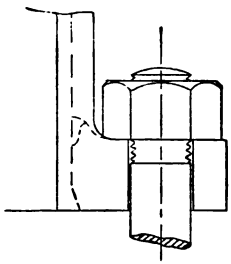


FIG. 47.

Table LXXXIII. (page 277) gives suitable thicknesses for bronze pipes, elbows, T pieces, &c., and will be of assistance in designing bronze stop-valves, expansion stuffing boxes, &c.

The basis and method of construction of the Table are fully explained in the following memoranda:—

Memoranda.—H.P. includes feed and bilge delivery, blow-off, scum, steam, fire service, and all sea valves.

L.P. includes all suctions, water pipes not under pressure, exhaust, and waste steam.

The necks of all boiler fittings should be $\frac{1}{16}$ inch thicker than H.P. list.

For plain pipes or cylinders the list is calculated for working pressures of 180 lbs. and 90 lbs. per square inch, for the H.P. and L.P. columns respectively; the stress per square inch of material is taken at 2240 lbs., and a constant addition of ·05 inch is made to cover errors in casting; the formulæ are therefore $t = ·08$ rads. + ·05 inch for H.P., and $t = ·04$ rads. + ·05 inch for L.P.

For H.P. T pieces the square of the thickness (before addition of ·05 inch) has been increased by 33 per cent.,—as a T pipe with all three branches of equal diameter is considered to be that amount weaker than the corresponding plain cylinder without branch in side.

No increase is made in the thickness of L.P. T pieces (over plain pipes), as the pressure assumed, viz. 90 lbs., is already high.

The B.M.E.D. & C. Committee deprecate the use of brazed copper pipes and recommend solid drawn of copper 99·25 and arsenic 0·25 to 0·45 per cent.; the hydraulic test pressure should stress the material to 7500 lbs.; no pipe for a working pressure over 180 lbs. should be of copper if above 6-inch bore. Hydraulic tests for steam pipes to be twice and for feed pipes 2·5 times the working pressure. If, when bent, the angle between the parts is less than 150°, the thickness must be 15 per cent. greater than by rule. The radius of curvature at centre line to be not less than two diameters.

When cast-metal pipes are used for branches, bends, &c., the following rule to be observed:—

$$\text{Thickness in 32nds inch} = \frac{\text{working pressure} \times D}{k} + x,$$

D is the bore in inches.

Table LXXXIII.—Thickness of Bronze and Cast-Steel (over ⅜) Pipes, T Pieces, &c.

Diameter in inches.	Plain pipes.				T pieces.			
	Calculated thickness.		Thickness to be.		Calculated thickness.		Thickness to be.	
	H.P.	L.P.	H.P.	L.P.	H.P.	L.P.	H.P.	L.P.
inches.	inch.	inch.	inch.	inch.	inch.	inch.	inch.	inch.
1	·09	·07	¼	¼	·10	·07	¼	¼
1¼	·10	...	¼	¼	·11	...	¼	¼
1½	·11	·08	¼	¼	·12	·08	¼	¼
1¾	·12	...	¼	¼	·13	...	¼	¼
2	·13	·09	¼	¼	·14	·09	¼	¼
2¼	·14	...	¼	¼	·16	...	¼	¼
2½	·15	·10	¼	¼	·17	·10	¼	¼
2¾	·16	...	¼	¼	·18	...	¼	¼
3	·17	·11	¼	¼	·19	·11	¼	¼
3¼	·18	...	¼	¼	·20	...	¼	¼
3½	·19	·12	5/16	¼	·21	·12	5/16	¼
3¾	·20	...	5/16	¼	·22	...	5/16	¼
4	·21	·13	5/16	¼	·23	·13	5/16	¼
4½	·23	·14	5/16	¼	·26	·14	5/16	¼
5	·25	·15	5/16	¼	·28	·15	5/16	¼
5½	·27	·16	⅜	5/16	·30	·16	⅜	5/16
6	·29	·17	⅜	5/16	·33	·17	⅜	5/16
6½	·31	·18	⅜	5/16	·35	·18	⅜	5/16
7	·33	·19	⅜	5/16	·37	·19	⅜	5/16
7½	·35	·20	⅜	5/16	·40	·20	7/16	5/16
8	·37	·21	⅜	5/16	·42	·21	7/16	5/16
8½	·39	·22	7/16	5/16	·44	·22	7/16	5/16
9	·41	·23	7/16	5/16	·46	·23	½	5/16
9½	·43	·24	7/16	5/16	·49	·24	½	5/16
10	·45	·25	½	5/16	·51	·25	9/16	5/16
10½	·47	·26	½	⅜	·53	·26	9/16	⅜
11	·49	·27	½	⅜	·56	·27	9/16	⅜
11½	·51	·28	9/16	⅜	·58	·28	⅝	⅜
12	·53	·29	9/16	⅜	·60	·29	⅝	⅜
12½	·55	·30	9/16	⅜	·63	·30	⅝	⅜
13	·57	·31	⅝	⅜	·65	·31	11/16	⅜
13½	·59	·32	⅝	⅜	·67	·32	11/16	⅜
14	·61	·33	⅝	⅜	·70	·33	¾	⅜
14½	·63	·34	11/16	⅜	·72	·34	¾	⅜
15	·65	·35	11/16	⅜	·74	·35	¾	⅜
15½	·67	·36	11/16	⅜	·76	·36	13/16	⅜
16	·69	·37	¾	⅜	·79	·37	13/16	⅜
16½	·71	·38	¾	7/16	·81	·38	13/16	7/16
17	·73	·39	¾	7/16	·83	·39	⅞	7/16
17½	·75	·40	¾	7/16	·86	·40	⅞	7/16
18	·77	·41	13/16	7/16	·88	·41	⅞	7/16
18½	·79	·42	13/16	7/16	·90	·42	15/16	7/16

For cast iron (tensile strength 9 tons), $k=200$, $x=6$.
,, cast steel (tensile 28/35 tons), $k=400$, $x=8$.
,, good bronze (tensile 15 tons), $k=225$, $x=4$.

The minimum thickness for cast metals in 32nds $=2\cdot5\sqrt{D}+z$. For cast iron z is 4, for steel 6, and for bronze 2.

Solid-drawn steel pipes can now be obtained up to 24 inches bore, and welded steel or iron pipes of any diameter in 100ths of an inch. The minimum thickness as prescribed by the B.M.E.D. & C. Committee for all wrought-iron or steel pipes is $5\sqrt{D}+2$. By this rule the limit for a 4-inch pipe is 0·12 inch, and for a 24-inch one 0·45 inch. Such pipes can be made considerably thinner, but with additional cost and difficulty in attaching the flanges, unless the ends are thickened by upsetting, as with stay tubes in boilers.

Welded steel or iron tubes are now also used for steam pipes, and provided they are made of a material which can be depended on to readily and certainly weld, there is no need for cover straps over the weld. For such a purpose steel with a tensile not exceeding 24 tons, with an extension not less than 30 per cent., and with a low sulphur content, should be used.

The flanges may be screwed on to the pipe or shrunk on and riveted; they may also be expanded into the flanges, having recesses turned on their inside to receive the impressed metal. Flanges are now often electrically welded to the pipes, and the welding is now so well done as to command confidence when proper materials are used.

The Board of Trade rules were as follows:—

170. The working pressure of well-made copper pipes is found as follows:—

$$\text{W.P.} = \frac{(t-3)}{D} \times F.$$

Where t is the thickness in 100ths of an inch.
D is the inside diameter in inches.
F for solid-drawn pipes is 60, for brazed 45.

All copper pipes used for steam, feed, blow-off or scum purposes subject to a pressure over 75 lbs. per square inch shall be solid-drawn, and when the working pressure exceeds 180 lbs. the bore shall not be more than 5 inches.

171. The internal pressure on lap-welded iron or approved steel pipes may be found as follows:—

(*a*) $$\text{W.P.} = \frac{(t-12)}{D} \times 90.$$

For pipes which are solid-drawn hot finished

(*b*) $$\text{W.P.} = \frac{(t-12)}{D} \times 120.$$

For pipes which have been solid-drawn cold finished

(*c*) $$\text{W.P.} = \frac{(t-10)}{D} \times 120.$$

All steel pipes should be efficiently annealed after being heated locally for welding or bending them to shape.

The flanges of wrought-iron and of steel pipes should be made of solid wrought material of ductile quality.

Pipes for superheated steam should be made of wrought-iron or steel, and not of copper.

172. Efficient means should be provided for draining all steam pipes. Boiler stop-valves cannot be regarded as suitable for this purpose. All drain cocks or valves should be accessible, and so placed as to render it practicable to drain the water from any portion of the steam pipes or chests in connection therewith. Drain pipes should be fitted to drain cocks or valves when the latter are in such a position that the water or steam discharged from them would be likely to cause personal injury. It is desirable that the drains should be automatic in their action.

173. The parts of a socket expansion joint subject to rubbing action should be made of brass or of other metal which will not rust.

In all cases in which such a joint is fitted to a bent steam pipe, the Surveyor should require a fixed gland and bolts, or other efficient means, to be provided to prevent the end of the pipe being forced out of the socket. This regulation should be complied with in all cases of bent pipes fitted with socket expansion joints, and it is also desirable that fixed glands and bolts should be fitted to the expansion joints of *straight* steam pipes, as cases have occurred, particularly with small straight pipes, in which the ends have been forced out of the sockets.

A socket expansion joint on a bent pipe is not a desirable arrangement, and, when adopted, the pipe should be anchored or provided with a strut at the bend, to relieve it of any undue bending stresses which might otherwise be produced by the internal pressure on a surface of the pipe equal to the area due to its bore.

174. Surveyors should pay particular attention to the examination and testing of steam pipes, and a record of the tests should be kept in the office boiler-book.

Tables LXXX. and LXXXI. are calculated in accordance with the rules given in paragraphs 170 and 171 of the above rules.

Pipe arrangements.—The greatest care should be taken in scheming arrangements of steam and exhaust pipes to keep them as far as possible in the same horizontal plane; a "pocket" in which water can collect must never be permitted under any circumstances. Neither should a steam pipe which has been led horizontally for some distance be suddenly bent up into the vertical; *all* sharp bends are sources of danger and to be avoided, but sharp rising bends are specially so.

The provision of proper arrangements for draining all steam pipes, regulator valve boxes, shut-off valve boxes, &c., is also a matter requiring the closest attention—as water that has once left the boilers and entered the steam pipes will never, with any ordinary arrangement of pipes, drain back to the boiler in the face of the issuing steam. Where the pipes are long it is very necessary to provide collectors of

some sort—as traps do not act with the necessary rapidity; the inlet pipes to the collectors should be large, and so placed as to arrest and lead away the rapidly moving water.

The question of the arrangements necessary to allow for the expansion of such pipes as vary much in temperature is a difficult one, and can only be very generally treated here.

Steam pipes for high pressures are so thick and rigid that ordinary bends are of little or no use for giving elasticity, and stuffing boxes should be provided wherever expansion and contraction will take place in ordinary work.

If expansion stuffing boxes and glands are not entirely of bronze, as is usual in Naval work, the glands should be of that metal, and the bodies so bushed or lined with it as to prevent the possibility of rusting up and consequent jamming.

These stuffing boxes should be cast with the necessary flanges or brackets to secure them to the bulkheads or other fixed bases, and the end of each length of piping (the end furthest away from the stuffing box) should be anchored in a similar way.

In the best class of work the end of the pipe that enters the stuffing box is always made of a separate piece in cast bronze, and this method has the advantage that the pair of flanges at the junction between the bronze pipe and the copper or iron one form a convenient "guard-flange," or flange to take the "guard" bolts, which prevent the pipe from being blown out of the stuffing box. These guard bolts should never be omitted except the pipe be very short and rigid, and the end attachments also very rigid—as fatal accidents have occurred in the past owing to their absence.

With the high pressures and temperatures now obtaining the faucet joint does not provide for the expansion of steam pipes inasmuch as if the packing is steamtight the friction is sufficient to grip the pipe. A more satisfactory arrangement is to have a carefully turned spigot whose end is bored so as to be thin enough to expand somewhat under pressure in the smoothly bored socket without any stuffing box and gland. This permits of the free expansion of the pipes without any leakage of steam or water. Both spigot and socket should be of incorrodable metals preferably of different compositions to avoid seizing.

It is also very important that those in charge of the fitting-up of the pipes on board ship should have full and clear information (a special small scale tracing or diagram is best) as to the amount of expansion anticipated, and to be provided for, at each stuffing box.

Steel Steam Pipes.

For steam pressures over 180 lbs. per square inch steel pipes are now generally used. Up to 6 inches diameter they should be solid drawn, and above that they may be welded; they are better, however, solid drawn, as generally supplied for Naval work, and though somewhat more costly they are much lighter.

Solid-drawn steel tubes, without seam or weld, are now made in all sizes up to 24 inches diameter, and can be obtained (Chesterfield Tube Co.) of a minimum thickness equal to 0·015 of the diameter; they are usually made of special steel having an ultimate tensile strength of about 27 tons, with an elongation of 35 per cent. in 2 inches; but they can be made of a stronger material if required. Such tubes are finished by cold drawing, and when so very thin are, of course, very expensive; but when, in accordance with the following rules, which hold good for steam and all other pipes exposed to pressure, the cost is moderate, especially considering their safety and lightness.

The diameter is D, the length L, both in inches; the pressure is P in pounds per square inch:—

Rule 235. Thickness of steam cold solid-drawn pipes in ins.

$$= \frac{D \times P}{12{,}000} + 0{\cdot}08,$$

or **in 64ths of an in.** $= \frac{D \times P}{185} + 5.$

Thickness of hot drawn in 64ths of an in.

$$= \frac{D \times P}{185} + 8.$$

Pipes of all sizes can be made from sheets by lap-welding the joints, and both with iron and mild steel a good and reliable tube is produced. There is, however, the uncertainty of the weld being continuous and complete, consequently the authorities cannot treat it on the same lines as a weldless one. Main steam, reduced steam, exhaust and other pipes are now made usually in the Mercantile Marine of welded steel without any cover strap. The working pressure can be ascertained by the Board of Trade Rule, p. 278, and by referring to Table LXXXI., p. 274.

Rule 236.

Thickness of welded, &c., steam pipes in 64ths of an in. $= \frac{D \times P}{140} + 7.$

Exhaust pipes subject to external pressure of 15 lbs., continuously or temporarily, may be solid-drawn, welded, or riveted of a thickness given by the following:—

Rule 237. The thickness of exhaust pipes within which the pressure may be less than the external, solid-drawn or welded.

$$\text{Thickness in 64ths of an inch} = \sqrt{\frac{D \times L}{6}} + 4.$$

L is the length of pipe between the flanges in inches.

Tables LXXXIV., LXXXV., and LXXXVI. give the thicknesses of the various pipes in 64ths of an inch, as they can be made, worked, and used in practice. The Admiralty have all such pipes galvanised by the cold process; as a preventive of corrosion, both inside and out, it is well worth the cost. No doubt the zinc will, in course of time, wear away from the inside, but it will always protect the outer surface.

Table LXXXIV.—Thickness of Cold Solid-drawn Steel Steam and Feed Pipes in 64ths of an inch.

Diameter of Bore.	Working pressures in lbs. per square inch.													
	50	60	70	80	100	120	140	160	180	200	220	240	260	280
ins.														
2·0	8	8	9	9	9	9	9	9	10	10	10	10	11	11
2·5	8	9	9	9	9	10	10	10	10	10	11	11	12	12
3·0	8	9	9	9	10	10	10	10	10	11	11	12	12	12
3·5	9	9	9	10	10	10	10	11	11	11	12	12	12	13
4·0	9	9	10	10	10	10	11	11	11	12	12	12	13	14
4·5	9	10	10	10	10	11	11	11	12	12	13	13	13	14
5·0	10	10	10	10	11	11	11	12	12	13	13	13	14	15
6·0	10	10	10	10	11	11	12	12	13	13	14	14	15	16
7·0	10	10	10	11	11	12	12	13	13	14	15	15	16	17
8·0	11	11	11	11	12	12	13	13	14	15	16	16	17	18
9·0	11	11	11	11	12	13	13	14	15	16	17	17	18	19
10	12	12	12	12	13	13	14	15	16	17	17	18	19	20
11	13	13	13	13	14	14	15	16	16	17	18	19	20	21
12	15	15	15	15	15	15	16	16	17	18	19	20	21	22
13	16	16	16	16	16	16	17	17	18	19	20	21	22	23
14	18	18	18	18	18	18	18	18	19	19	21	22	23	24
15	19	19	19	19	19	19	19	19	20	20	21	22	24	24
16	20	20	20	20	20	20	20	20	21	21	22	23	24	25

N.B.—The pressure in feed pipes will be greater than the boiler pressure, and if from pumps worked by the main engines, should be taken at 1·3 to 1·6 times the working pressure. If from independently worked direct pumps, 1·2 to 1·4 times is ample.

Table LXXXV.—Thickness of Welded or Riveted Steam Pipes in 64ths of an inch.

Diameter of Bore.	Working pressure in lbs. per square inch.															
	50	60	70	80	90	100	120	140	160	180	200	220	240	260	280	300
ins.																
8	13	13	13	14	14	14	15	15	16	16	17	17	18	18	19	19
9	13	13	13	14	14	14	15	15	16	17	17	18	18	19	19	20
10	13	13	14	14	14	15	15	16	17	17	18	18	19	20	20	21
11	13	13	14	14	15	15	16	16	17	18	18	19	20	21	21	22
12	13	14	14	15	15	15	16	17	18	18	19	20	21	22	22	23
13	13	14	14	15	15	16	16	17	18	19	20	20	21	22	23	24
14	13	14	15	15	16	16	17	18	19	19	20	21	22	23	24	25
15	14	14	15	15	16	16	17	18	19	20	21	22	23	24	25	26
16	14	15	15	16	16	17	18	19	20	21	22	23	24	25	26	27
17	14	15	15	16	16	17	18	19	20	21	22	23	25	26	27	28
18	14	15	16	16	17	17	18	20	21	22	23	24	25	27	28	29
19	15	15	16	16	17	18	19	20	21	22	24	25	26	28	29	30
20	15	15	16	17	17	18	19	21	22	23	24	25	27	28	30	31
21	15	16	16	17	18	18	20	21	22	24	25	26	28	29	31	32
22	15	16	17	17	18	19	20	22	23	24	25	27	29	30	32	32
23	15	16	17	17	18	19	20	22	23	25	26	28	29	30	32	32
24	15	16	17	18	19	20	21	23	24	25	26	28	30	31	32	32

Table LXXXVI.—Thickness of Exhaust Pipes (Steel), Solid-drawn or Welded, in 64ths of an inch.

Diameter of Bore.	Length of pipe in inches between flanges or ribs.													
	60	65	70	75	80	85	90	95	100	110	120	130	140	150
ins.														
6	7	7	8	8	8	8	8	8	8	9	9	9	9	9
7	7	8	8	8	8	9	9	9	9	9	9	9	10	10
8	8	8	8	9	9	9	9	9	9	9	9	10	10	10
9	8	8	9	9	9	9	9	9	9	10	10	10	10	10
10	8	8	9	9	9	9	9	9	10	10	10	10	11	11
11	8	9	9	9	9	9	9	10	10	10	10	11	11	11
12	9	9	9	9	9	9	10	10	10	10	11	11	11	11
13	9	9	9	9	10	10	10	10	10	11	11	11	11	12
14	9	9	10	10	10	10	10	10	11	11	11	11	12	12
15	9	9	10	10	10	10	10	11	11	11	11	12	12	12
16	10	10	10	10	10	10	11	11	11	11	12	12	12	12
17	10	10	10	10	10	11	11	11	11	12	12	12	12	13
18	10	10	10	10	11	11	11	11	11	12	12	12	12	13
19	10	10	11	11	11	11	11	11	12	12	12	12	13	13
20	10	10	11	11	11	11	12	12	12	12	12	13	13	13
22	10	11	11	11	11	12	12	12	12	13	13	13	13	14
24	11	11	11	11	12	12	12	12	13	13	13	14	14	14
26	11	11	11	12	12	12	12	13	13	13	14	14	14	15
28	11	11	12	12	12	12	13	13	13	14	14	14	15	15
30	11	12	12	12	12	13	13	13	13	14	14	15	15	15
32	12	12	12	12	13	13	13	13	14	14	15	15	15	16
34	12	12	13	13	13	13	14	14	14	15	15	15	16	16
36	12	13	13	13	13	14	14	14	14	15	15	16	16	17

The thicknesses are for straight pipes only; bends to be suitably strengthened.

Welded pipes should be subjected to a water-pressure test of about three times the working pressure, and should be smartly struck with a steel hand hammer all along the weld while the pressure is on. The Admiralty at present require all welded pipes to be fitted with an external single-riveted butt-strap over the weld, but some of our most experienced tube makers are more than doubtful of the value of this addition.

Solid-drawn pipes up to about 5 inches diameter may be readily bent to moderate curves by filling with lead and bending cold in a press, as is usual with copper pipes. Lap-welded iron pipes of 3, 4, and 5 inches diameter, galvanised externally, are largely used in H.M. Navy for bilge suctions and connections, and are bent by filling with sand and heating to a low red heat. The suitable thickness in 64ths of an inch is 4 + 2 for each inch of bore.

The flanges for steel pipes should be stamped from steel plates, and may be made with a small bead on the back for such sizes as are screwed on to the pipes, and of angle bar section for those that are to be riveted on. For the smaller sizes, say up to 5-inch pipes, a good plan is to cut a half thread on the inside of the flange and then expand the pipe against the surface so grooved.

Tests of Steam Pipes (Solid-drawn Steel).

The present Admiralty requirements as regards steel steam pipes are as follows:—

To be made from acid open-hearth steel, which must stand the following tests:—

Annealed strips cut lengthwise from each pipe must have a tensile strength of between 24 and 27 tons, and must show an elongation of 33 per cent. in 2 inches before fracture.

Strips cut either way, 1½ inches wide, heated to a low cherry red and quenched in water at 82° F., must stand bending double in a press, over a radius equal to 1½ times thickness of strip tested, without fracture.

Pieces 2 inches long cut from ends of pipes must stand hammering down endwise, when cold, until length is reduced to 1 inch, and flattening until sides are within twice the thickness of the material of each other, without signs of fracture. Pipes intended to carry 300 lbs. steam must also, after pickling, and after fitting and facing of flanges, but before galvanising externally, stand satisfactorily a water-pressure test of 600 lbs. per square inch.

Steam Pipes (Welded Steel).

To be made from acid open-hearth steel.

All welding to be by gas on a mandril.

Each welded joint to be covered by an external single-riveted butt-strap at least $\frac{1}{16}$ inch thicker than pipe.

Edges of strap to be bevelled to an angle of 75°.

Tensile and bending tests as for solid-drawn pipes (above).

The plates are, in addition, to stand such other forge tests, hot and cold, as the overseer may consider necessary, and pieces are to be welded and broken in the testing machine; also some pieces cut from the pipes are to be broken to ascertain the efficiency of the welding.

Pipes of this construction must also stand the water-pressure test specified above for solid-drawn pipes.

Flanges of Steel Steam Pipes.

To be forged from the solid, not welded.

To be made from acid open-hearth steel.

Material to have a tensile strength of between 27 and 30 tons per square inch, and to show 27 per cent. elongation in 2 inches before fracture.

Bending tests as for material of pipes (above).

STOP-VALVES.

These should always be made with the spindle a separate piece, distinct from the valve, and so attached as to allow a little "play,"—thus enabling the valve to accommodate itself to the seat when the latter is slightly "out of truth."

Steam stop-valves, above about 3½ inches diameter, should have external and accessible spindle nuts, carried in crossheads or bridges; for water valves this is not so necessary, but it is usually best to make only one type of valve for both purposes,—only varying the strengths of the parts.

Screw glands should not be used for valves over 1 inch in diameter, and, when used, should always be fitted with some means of locking.

Ordinary gland studs should be fitted with "check" or "lock" nuts and split pins.

Hand-wheels for stop-valves should never be turned and polished; they are better left rough and served with twine.

For large valves (over 10 inches in diameter, say) a forged cross-handle (four arms), with the ends turned over at right angles, either towards or away from the valve, is better than a wheel; many engineers prefer this type of handle to the hand-wheel for all valves above about 4 inches diameter, and there is much to be said in its favour.

For boiler-room valves a similar forged handle, but with only two arms, one bent outwards and the other straight, is the best.

For very large stop-valve chests the best material is, undoubtedly, cast-steel. Trouble is occasionally caused by unsound castings, but this cannot be regarded as unavoidable.

In Naval practice, all stop-valves for water, oil, air, &c., are entirely of bronze, and steam valves of small and moderate sizes are constructed

in the same way, whilst the larger sizes have commonly cast-steel chests with bronze seats, &c.

The number of joints may often be considerably reduced by casting the expansion stuffing boxes along with the stop-valve chests.

The handling of large shut-off valves may be much facilitated by the addition of a small by-pass valve.

The whole of the valves in any vessel should open or close by turning the hand-wheel in the same direction. The Admiralty rule is to close with a right-hand motion. To remove all doubt, it is a good plan to cast on the upper surface of all hand-wheels the words "open" and "shut," thus:—OPEN←X→SHUT.

THE BALANCING OF ENGINES.

In an unbalanced engine there are two perfectly distinct sets of forces which give trouble; first, those set up by unbalanced rotating parts; and second, those caused by unbalanced reciprocating parts. The effect of an unbalanced rotating part, such us a crank, is to produce a force acting radially from the shaft axis through the centre of gravity of the part, which causes the shaft to spring and its axis to revolve in a small orbit round the centre of rotation, the shaft thus moving eccentrically and tending to grind out the bearing round its whole circumference. This may result in undue wear and tear and heating of bearings; if the bearing surface is ample, the force will tend to spring the engine seatings and cause the whole engine to move with the eccentrically-moving shaft; if the seatings are rigid, the whole hull of the ship may respond to the movement of the engines and thus be caused to vibrate.

The forces set up by vertically reciprocating parts are mostly vertical, and, if their resultant acts through the centre of gravity of the engine, the tendency is to *tilt* the whole engine and ship up and down, but if acting away from that vertical line of the engines they tend to rock the engines. The result in either case, if the vibrations are severe or the hull is lightly built, is to cause deterioration of the ship's structure, with more or less leakage, and also discomfort to passengers and crew, and in the case of warships to produce faulty gunnery. Even small unbalanced forces may produce great vibration if their periods synchronise with the vibration periods of the hull or any part of it.

Part of the connecting-rod should be considered as reciprocating with the piston, &c., and the other part as revolving with the crank-pin, &c., and the relative magnitudes of these parts may be determined as follows:—Find the centre of gravity of the rod, and if it be found to lie at, say, 70 per cent. of the rod's length from the top end, the reciprocating weight may be taken as 100 – 70, or 30 per cent. of the total weight of rod; and the remaining 70 per cent. must then be treated as revolving with the crank-pin, &c.

Strictly speaking, the force set up by a rotating weight can only be truly balanced, or neutralised, by the similar and equal force set up by

another weight placed at the opposite side of the axis and rotating in the same transverse plane; and the force set up by a reciprocating weight can only be neutralised by the similar and equal force set up by another weight reciprocating on the same line, but in the opposite direction.

A single crank-arm balance weight cannot be arranged to rotate with its centre of gravity in the transverse plane through the mid-length of the crank-pin; it is therefore halved and placed one on each arm, symmetrically. Strictly, such balance weights should not have a greater moment (weight × radius of centre of gravity) than the unbalanced rotating parts, for if they are made heavier, with a view to balance also a portion of the reciprocating weights, a horizontal force is set up equal in magnitude to that portion of the vertical inertia force that is balanced.

Still there are cases, such as those of the numerous auxiliary engines in which the reduction of the inertia forces of the reciprocating parts, and the vibration caused thereby, are of greater importance than the creation of small transverse forces; hence, for the generality of such cases this simple method of "overbalancing" serves admirably. A very small weight attached to the rim of a fan, or placed in the crank-disc of a centrifugal pump engine, though not in the transverse plane through the mid-length of crank-pin, yet produces a marked improvement in the running of the single-crank engine, and should be always fitted unless a more perfect arrangement is necessary. Balance weights forged with the crank-arms are, of course, better, but, except in the case of built shafts, they are very much more expensive.

A piston, with rod, &c., cannot usually be balanced by another weight reciprocating in the opposite sense in the same vertical line, but "bob-weights," driven by special eccentrics or cranks, can be symmetrically arranged as regards the transverse plane through the centre-line of piston-rod, and may be on the same side of the crankshaft as the piston, &c., to be balanced; but the extra cost, weight, and complication are serious objections, and have led to the perfecting of what may be called "collective" balancing—a method which secures to the full the advantage of quenching vibration, though leaving the wear and tear of bearings, to some extent at any rate, to be resisted by ample surfaces.

As already remarked, the force set up by an unbalanced rotating weight is always radial and acts from the shaft axis through the centre of gravity of the rotating weight. Its magnitude (neglecting gravity) is given by :—$F = WrN^2 \times \cdot00034$,* where W = weight of body in pounds, r = radius of its centre of gravity in feet, and N = number of revolutions per minute. The vertical component of this force for any crank position has the value :—$F_V = WrN^2 \times \cdot00034 \times \cos\theta$, where θ is the angle between crank and engine centre-line; or, if a graphic method is preferred, for $\cos\theta$ read $\frac{AB}{AC}$ as indicated in Fig. 48.

* Obtained by substituting $\frac{2\pi r}{60}N$ for V in the formula $F = \frac{WV^2}{gr}$.

This fraction has the value 1 at the top and bottom centres, 0 at the points D and E, and intermediate values at other positions.

Similarly, the horizontal component is:—$F_H = WrN^2 \times \cdot00034 \times \sin\theta$, or, graphically, for $\sin\theta$ read $\frac{BC}{AC}$. This fraction has the value 1 at positions D and E, and 0 at top and bottom centres.

The force due to an unbalanced weight reciprocating vertically is purely vertical, as previously stated; and it is a maximum at each end of the stroke and falls to zero at a point near mid stroke. Its magnitude and its effect upon the effective pressure diagram (fig. 2), may be determined by Mr Rigg's method as follows:—Describe a circle (fig. 49) with radius equal to $WrN^2 \times \cdot00034$, to any convenient scale, W being in this case the weight in pounds of the reciprocating parts, which are

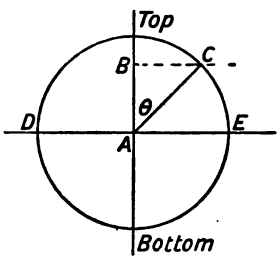


FIG. 48.

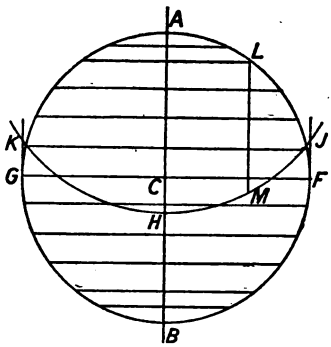


FIG. 49.

here considered as being located at the crank-pin and revolving with it. Divide the diameter AB into ten equal parts and re-divide each end part into two. Draw horizontal lines through these division marks to meet the circle as shown. Then—supposing the connecting-rod to be four cranks in length—from C set off CH equal to one-fourth CB; and from F and G set off, in the opposite direction, FJ and GK, each equal to CH; and through J, H, and K strike a circular arc (in this case from point A). Then the distances LM, &c., will give the forces required at each of the thirteen (including the two end points) points in the stroke, and they may be designated − or + according as they are measured above or below arc JHK.

Draw a horizontal line AB (fig. 50) to represent the stroke of the piston, and divide it similarly to AB in the previous figure, and at each point set off, up or down, the force given by the ordinate of that figure, divided by the piston area. Then, if a fair curve be drawn through the ends of these ordinates, it will show the variation in the

pressure per square inch required to accelerate and retard the piston, &c., during one stroke, if gravity be neglected. The variation during the return stroke is shown by the dotted line. But gravity must be allowed for, and to do this a new base line must be drawn at a distance *w* above AB (or *below* AB for the up stroke), *w* representing to scale the weight of the reciprocating parts per square inch of piston,—and all measurements must then be taken from *ab* in lieu of AB. Now draw the effective pressure diagram (fig. 2), above this figure, and to the same scale, and add to, or subtract from, each of its ordinates the amount indicated by the lower figure; and, through the new points thus found, draw a fair curve as indicated by the dotted line CED. The ordinates from CD to this curve will give the net (except as regards friction losses) effective pressures acting through the piston rods; and

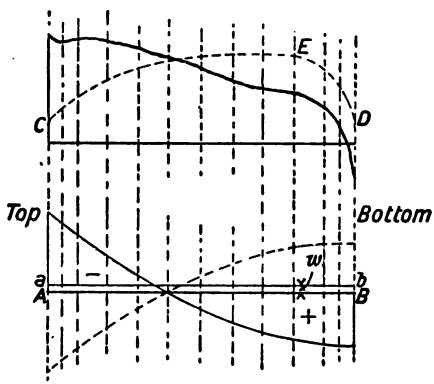


FIG. 50.

from these the net pressures at the crank-pin, or the net twisting moments, may be determined by the method shown in fig. 15, and may be exhibited by a diagram resembling fig. 16, or by the circular form of the same diagram. The ordinates of these figures will then give the net crank-pin pressures, or the net twisting moments, at every point in the crank-pin path, except as regards friction losses and the small fly-wheel effect of the propeller, cranks, &c.

The sum of the unbalanced forces set up by any crank and its piston, piston-rod, connecting-rod, &c., at every position during a revolution, may be exhibited by a diagram constructed in the following manner:—

First, find the amount of the radial force set up by the rotating parts, by the method already given, and describe a circle with a radius representing this force to any convenient scale (fig. 51). And secondly, from a selected number of points in the circumference of this

circle (which is also to be considered as representing the path of the crank-pin) set off, upwards or downwards, vertical ordinates representing, to the same scale, the forces due to the reciprocating parts (from fig. 65), and through the extremities of these ordinates draw a fair curve. Then for any crank angle, say 30° down from top centre, the total force is indicated in magnitude and direction, by the resultant CB of the two forces CA and AB, and so for other positions of crank.

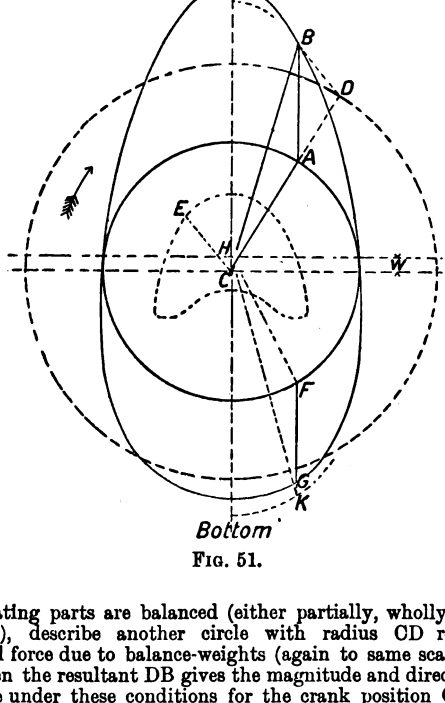


FIG. 51.

If the rotating parts are balanced (either partially, wholly, or "over-balanced"), describe another circle with radius CD representing centrifugal force due to balance-weights (again to same scale) and join DB. Then the resultant DB gives the magnitude and direction of the total force under these conditions for the crank position CD; and if lines such as CE, equal and parallel to the resultants so found, be set off from the centre C, and a fair curve be drawn through their extremities, the new diagram thus produced shows at a glance the variations in the total force during one complete revolution. Note that in this case the lines CE, &c., are set off the reverse way round.

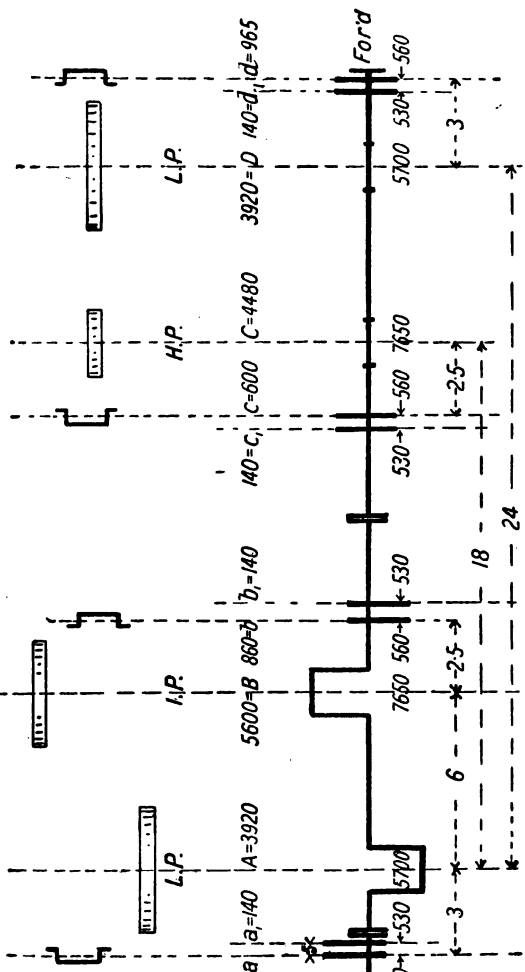

965=a
a₁=140
A=3920
5600=B
860=b
b₁=140
140=c₁
c=600
C=4480
3920=D
140=d₁
d=965
L.P.
I.P.
H.P.
L.P.
Aft.
Ford
560
530
5700
7650
560
530
560
530
7650
560
530
5700
530
560
3
6
2·5
18
24
2·5
3

FIG. 52.

If gravity is to be allowed for, as it must be when estimating forces acting on main bearings, framings, &c., a new horizontal centre line should be drawn at a distance W above the first one,—where W represents the weight of the rotating parts, to scale; and the ordinates AB, FG, &c., must be respectively shortened and lengthened by an amount representing the weight of the reciprocating parts. All measurements should then be taken from the point H, and HK would be one of the set of resultants corresponding to CB, &c., in the former case, where gravity is not considered.

When vibrations and the forces causing them are alone under consideration, gravity need not be considered, for the reason that the weights of the moving parts may be regarded as producing a slight initial depression of the engine seatings, &c., in addition to that caused by the weights of the stationary parts, and the vertical vibrations may then be considered as taking place equally above and below this lowered position, instead of unequally above and below the higher one.

The object of the "collective" system of balancing (best known as the "Yarrow, Schlick, and Tweedy System") is not to balance the moving parts in connection with each cylinder independently, but so to space the centres of the various cylinders, so to arrange the various crank angles, and so to proportion the weights of the moving parts as to cause the whole of the disturbing forces to cancel out. In order that an engine may be balanced on this system, it must have not less than four cranks, and, generally speaking, engines with fewer cranks must be balanced by some combination of the rotating balance weight and the reciprocating "bob-weight."

As soon as the general design and arrangement and leading dimensions of an engine (say four-crank triple) have been decided upon, its condition as regards balance may be examined by the following approximate method, approximate because it takes no account of disturbances due to obliquity of connecting rod:—

Make a skeleton sketch of the engine (as indicated by fig. 52) and figure upon it the weights of the unbalanced rotating parts referred to the crank-pin $\left(i.e.\ \frac{\text{Weight of part} \times \text{dist. of its c. of g. from shaft axis}}{\text{Radius of crank path}}\right)$, the weights of the reciprocating parts, and the distance of each set of parts from an imaginary transverse plane of reference through the mid-length of the after crank-pin. Then, to examine the forces and moments due to the reciprocating parts, which act in the vertical plane through the shaft axis and tend to set up vertical vibrations, first tabulate the data thus:—

Table LXXXVIa.

Marks on Cranks, etc.	Weights of Recipro-cating Parts. lbs.	Distances from Ref. Plane to Centres of Piston-rods, etc. ft.	Moments of Recipro-cating Parts. ft. lbs.	Weights of Unbalanced Rotating Parts. lbs.	Moments of Rotating Parts. ft. lbs.
a	965	−3	−2,895	560	−1,680
a_1	140	−2·5	− 350	530	−1,400
A	3,920	...	...	5,700	...
B	5,600	6	33,600	7,650	45,900
b	860	8·5	7,310	560	4,760
b_1	140	9	1,260	530	4,770
c_1	140	15	2,100	530	7,950
c	600	15·5	9,300	560	8,680
C	4,480	18	80,640	7,650	137,700
D	3,920	24	94,140	5,700	136,800
d_1	140	26·5	3,710	530	14,045
d	965	27	26,055	560	15,120

In this particular case the stroke of pistons was five times the travel of valves; the weights of valve and gear given in columns two and five are therefore one-fifth of the actual weight. The minus sign is used because the after low-pressure gear is to the left of the reference plane, whilst all other weights are to the right of it. The weights in connection with the after low-pressure crank have no moment about the plane of reference, because the plane passes through their centre of gravity.

As regards valve gear, the valve, rod, block, half the links and drag rods, and a proportion of the eccentric rod and strap (say, four-tenths) are taken as reciprocating over the ahead eccentric; and the remaining half of the links, and a similar proportion of the astern eccentric rod and strap, over the astern eccentric. The unbalanced portions of the sheaves and the remainders of the rods and straps (six-tenths) are taken as rotating parts.

Additions are made to the reciprocating weights to allow for the unbalanced parts of pump levers, links, rods, and buckets.

The obliquity of eccentric rods is usually so small that it may be neglected.

Next assume (as first working hypothesis) that the cranks are arranged with a forward pair, opposite to one another, and an after pair, also opposite to one another, but that the forward pair are at right angles to the after pair. This combination has the advantage of getting the lighter L.P. engines (with the reduced rods, etc.—L.P. powers being usually about five-eighths of H.P. or I.P. powers) at the

ends, and also permits them to take steam alternately. If the engine is a four-crank quadruple it is better to make the heavier L. P. one of the middle cylinders and keep the lighter ones at the ends.

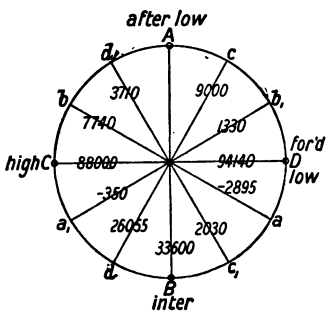


FIG. 53.

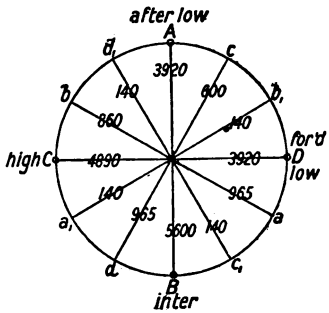


FIG. 54.

Now describe two circles (figs. 53 and 54), each representing the crank path ; draw in the various cranks and eccentrics at their correct angles, and figure the moments on one and the forces on the other, as

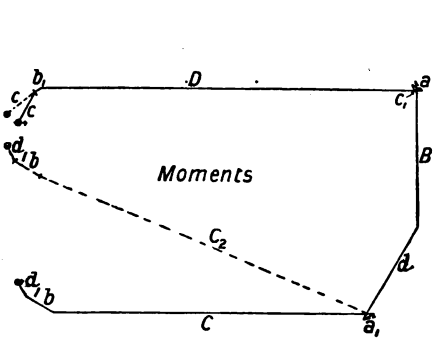


FIG. 55.

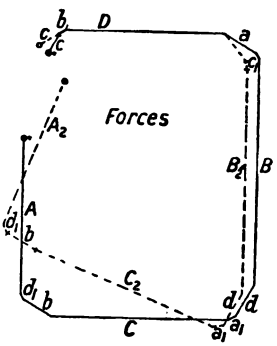


FIG. 56.

indicated. Then draw a polygon (fig. 55) in which the sides are successively parallel to the radial lines in fig. 53, and represent, to some convenient scale, the moments figured on them. Also draw a similar polygon (fig. 56) for the forces, from fig. 54).

If these polygons are closed figures, the moments and forces will be in equilibrium ; but if they do not close, the additional side required

to complete closure will represent, in magnitude and direction, the moment or force required to produce equilibrium. There are four principal weights, three principal distances (from centre to centre of cylinders), and four crank angles, or eleven principal variables, any one of which may be altered in moderation, and every alteration to which will have a material effect on the balance of the whole: and as the polygons (shown in full lines) do not close, it must now be decided what alterations shall be made to obtain the desired perfection of balance. As there are only three principal moments (viz., those of the piston, &c.), it is evident that the moment polygon, to close, must be roughly triangular in shape, and this shows that some alteration in crank angles will be necessary. In making such alteration it is well to bear in mind the interchangeability of the crankshafts—supposing that, as is usual, there are two of these, each carrying two cranks. To meet this requirement the angle between A and B should be the same as that between C and D; then the forward half, when reversed, may be placed in the after bearings in case of need, and *vice versâ*. The reversal is only necessary when the L.P. crank-pins are shorter than the others, as has been assumed in this case.

The dotted lines on figs. 55 and 56 show one solution of the problem. The cranks C and A are each moved 22½° ahead and the weight of the H.P. piston is increased by 410 lbs.—the figures then practically closing. It must be noted that when a crank angle is altered in these polygons the angular position of its eccentrics must also be altered; and before such an alteration is decided upon, the effect it will have should be examined both in the moment and in the force diagrams.

Similar polygons should now be constructed for the unbalanced rotating weights (which, by the way, if balanced in one plane are also in balance in any other plane through the shaft axis); and if the same crank angles, &c., do not appear to be the most suitable, some splitting of differences must be resorted to. As a rule, the weights of the rotating parts bear such a relation to those of the reciprocating parts that the same crank angles, &c., will be found to suit both.

It will, of course, be apparent that, with so many variables, more than one solution may be quite practicable.

When the leading features of a design have been worked out and settled by the methods outlined above, the results, as regards any residual unbalanced forces, may be exhibited upon a diagram such as that shown in fig. 57, which is constructed as follows:—

First, take a straight line PQ to represent the axis of the crankshaft, and lay off from it, at their proper fore and aft positions, ordinates (such as CC', DD', &c.) representing in magnitude and direction the total vertical force (*i.e.* force due to reciprocating parts plus vertical component of force due to rotating parts) due to each item at, say, 0° position (top centre) of high-pressure crank; and also show the vertical centre line, MM, through the middle point of the whole engine (half way between outer cranks). Then collect all the pluses and all the minuses, and subtract from the greater sum the less to find the resultant force R,—to which affix its proper sign. Then, to

determine the fore and aft position of R, assume a reference plane at XX_1 (say), and find the moment of each force about it by multiplying the force (in lbs.) by the distance of its line of action (in feet) from XX_1, taking care to distinguish the products by their proper signs. And finally, subtract for resultant moment, and divide latter by resultant force R, when the dividend will be the distance, *y*, at which this force acts from XX_1; and a further subtraction, the position of MM being known, will give Z. In the notation used, plus moments act "clockwise," and minus moments "anti-clockwise." The conclusion is, then, that at 0° H.P. crank angle, there exists a couple of moment RZ tending to tilt up the forward end of the engines, on the after end of the bed-plate as fulcrum, and a vertical force R, acting on MM and tending to press the engines as a whole down upon their seatings.

In determining the vertical component of the centrifugal force due to the rotating parts, the angle of each crank or eccentric should be stated as the number of degrees between it and the nearest vertical radius, and will, therefore, never exceed 90°; and, using the same notation as in fig. 50, the sign round the upper 180° will be minus (upward force), and round the lower 180°, plus (downward force).

As regards the reciprocating parts, the radius of fig. 49 may be made to represent the centrifugal force due to 1000 lbs., and when the radial lines have been drawn at the correct angles and the acceleration forces (LM, &c.) determined, the latter may be multiplied by the number of thousands, or by the decimal of a thousand, requisite to give the total force in each case.

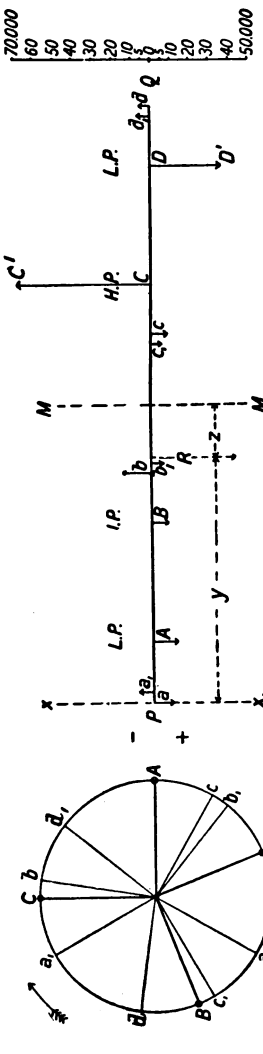


FIG. 57.

The moment RZ and the force R may now be represented by ordinates set up at 0° (fig. 58), and when similar ordinates have been determined for other crank positions (*e.g.* CD at 120°), curves may be drawn through their extremities, as in the figure, and these show at a glance the variations in the tilting moment and force during one complete revolution. It will, of course, be seen that if the engines were perfectly balanced these curves would become straight lines and would coincide with the base line; but this is a state of things rarely or never reached in practice, as, to mention one point only, estimated weights cannot be expected to tally exactly with the final weights of the parts, which have probably been altered in many points of detail since the

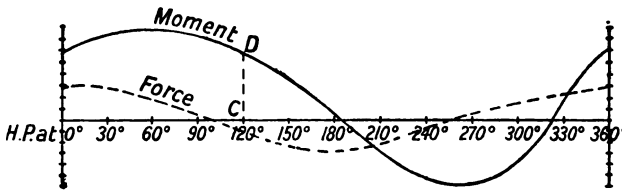


FIG. 58.

first design—on which the balancing calculations were based—was got out.

Similar curves may be constructed to show the residual moments and forces in the horizontal plane.

To give a clear idea of the condition of any set of engines as regards balance three diagrams should be presented, viz. :—

(*a*) Curves of residual vertical forces and moments.

(*b*) Curves of residual horizontal forces and moments.

(*c*) Curve of twisting moment.

And each diagram should have a scale of tons, or foot-tons, marked upon it; and the twisting moment diagram should also show the greatest and least moment as percentages of the mean. Three such diagrams were latterly required by the British Admiralty with each new design of a reciprocating engine submitted by a contractor.

Amongst other causes of vibration of minor importance may be mentioned—(*a*) the pendulum-like swing of the connecting-rod; (*b*) the thrusts on the piston-rod guides; and (*c*) irregular action of the propeller, due to disturbance of the water supply caused by the hull and fittings.

Every propeller should, as a matter of course, be accurately balanced.

BOILERS.

Fuels, Combustion, &c.

Coal.—The chief varieties of coal and their leading characteristics are as follows:—

(1) *Anthracite*, consisting almost entirely of free carbon, generally jet black in appearance, but sometimes greyish like plumbago, has a specific gravity generally of about 1·5, but sometimes as high as 1·9; it burns without emitting flame or smoke, but requires a strong draught to burn at all. It is capable of evaporating (theoretically) nearly 16 times its weight of water, but to obtain good results from it careful stoking is necessary, as when suddenly exposed to heat it does not cake but becomes very friable, breaks up into small pieces, and falls through the bar-spaces if disturbed much. The fires should be worked light when using it, and the coal carefully spread. The heat is very intense and local, so that furnaces intended to burn it should be high in the crowns and protected at the sides.

(2) *Dry bituminous coal* contains from 70 to 80 per cent. of carbon, and about 15 per cent. of volatilizable matter; its specific gravity is from 1·3 to 1·45. It burns easily and swells considerably while being converted into coke. The harder kinds do not burn so readily, nor do the pieces stick together so easily when burning, and are generally well adapted for marine boilers.

(3) *Bituminous caking coal*, containing from 50 to 60 per cent. of carbon, is generally of about the same specific gravity as the dry bituminous; it contains, however, as much as 30 per cent. of volatilizable matter, and consequently develops hydrocarbon gases; it burns with a long flame, and sticks together in caking, so as to lose all trace of the original forms of the pieces. It requires special means to prevent smoke.

(4) *Cannel coal*, or *long-flaming coal.*—This is seldom used for steam purposes, as it gives off large quantities of smoke, and is very scarce. It is the best coal for the manufacture of gas.

(5) *Lignite*, or *brown coal*, is of later formation than the other coals, and in some instances approaches to a peaty nature. It contains, however, when good, from 56 to 76 per cent. of carbon, and has a specific gravity from 1·20 to 1·35. It also contains large quantities of oxygen, and a small quantity of hydrogen. The commoner kinds of lignite are poor, and contain as little as 27 per cent. of carbon, and therefore are not suitable for steaming purposes.

Wood.—Dry wood contains on an average about 50 per cent. of carbon, 41 of oxygen, and 6 of hydrogen.

Patent fuels.—These usually consist of coal-dust, mixed with a little coal-tar and pressed into hard bricks; their value depends very much on the quality of coal from which they are made.

The value of a fuel is determined by its chemical composition. All fuels contain more or less carbon, most have also hydrogen and

oxygen in various proportions, and some contain small quantities of nitrogen, sulphur, &c., in addition.

The ordinary symbols for these substances and their combining weights are as follows:—

Carbon	— symbol	C	— combining weight	12.
Hydrogen	,,	H	,,	1.
Oxygen	,,	O	,,	16.
Nitrogen	,,	N	,,	14.
Sulphur	,,	S	,,	32.

When they combine chemically one with another, as in combustion, they invariably do so in the ratio of their combining weights, or of some multiple of the same. Thus,—

Carbonic oxide	— symbol	$CO = C_{12} + O_{16}$.
Carbonic dioxide	,,	$CO_2 = C_{12} + O_{32}$.
Water . .	,,	$H_2O = H_2 + O_{16}$.
Olefiant Gas .	,,	$CH_2 = C_{12} + H_2$.
Marsh gas .	,,	$CH_4 = C_{12} + H_4$.

Atmospheric air when dry is a mechanical mixture of nitrogen and oxygen in the proportion of 77 parts by weight of the former to 23 of the latter.

The heat developed by any substance during combustion is called its *total heat of combustion*, and is measured in *units of heat* or *thermal units*.

The British thermal unit is the amount of heat required to raise the temperature of one pound of pure water one degree Fahrenheit when at or near its greatest density (39·1° F.).

The mechanical equivalent of heat is the number of foot-pounds of energy required to raise the temperature of one pound of water one degree Fahrenheit; or in other words,—

1 British thermal unit* = 778 foot-pounds,
or 1 Horse-power = 33,000 foot-pounds = 42·42 thermal units.

To burn fully 1 lb. of carbon or convert it into carbon dioxide ("carbonic acid"), 2·7 lbs. of oxygen, or 12 lbs. of air are required, and a total heat of combustion of 14,500 units is developed,—*i.e.* sufficient to evaporate 15 lbs. of water from and at 212°. If, however, the air supply is restricted or deficient, there is a tendency to the production of carbon monoxide or "carbonic oxide,"—a product requiring only half the amount of oxygen for its formation,—and the total heat developed is then only 4400 units.

Each pound of hydrogen requires 8 lbs. of oxygen or 36 lbs. of air for its complete combustion, and develops in combining 62,032 units of heat,—sufficient to evaporate 64 lbs. of water from and at 212°.

* Joule's *mechanical equivalent*, 772, was for many years accepted as B.T. unit, but later experiments of a more refined nature have proved it to be 778 ft.-lbs.

Sulphur exists only in small quantities in good coal, and its total heat of combustion is only about 4000 units per pound.

When two substances combine chemically, as in combustion, the weight of the products of combustion is the sum of the weights before combination,—*e.g.*, 1 lb. of carbon uniting with 12 lbs. of air gives 3·7 lbs. of carbonic acid diluted by 9·3 lbs. of nitrogen; and the temperature of the products is found by dividing the total heat of combustion by their weight multiplied by their specific heat.

The specific heat of any substance is the quantity of heat required to raise the temperature of a pound of it one degree Fahrenheit, and is measured by the ratio that this quantity bears to the quantity required to raise the temperature of a pound of water one degree,—*i.e.* to the British thermal unit; thus, the specific heat of hydrogen, at a pressure of 30 inches of mercury, is 3·4, that of carbonic acid ·216.

When oxygen and hydrogen are both present in a fuel they are in combination as water and become steam without developing any heat; it is therefore only when there is an *excess* of hydrogen that any heating effect is produced. The only effect of nitrogen is to reduce the intensity of combustion and lower temperature of products.

The following rules are deduced from the above considerations:—

Rule 238. Total heat of combustion of one pound of fuel,—in thermal units. $= 14{,}500\left\{C + 4{\cdot}28\left(H - \frac{O}{8}\right)\right\}$.

Rule 239. Theoretical evaporative power of one pound of fuel,—expressed in pounds of water. $= 15\left\{C + 4{\cdot}28\left(H - \frac{O}{8}\right)\right\}$.*

Rule 240. Number of pounds of air required to burn one pound of fuel. $= 12C + 36\left(H - \frac{O}{8}\right)$.

Ordinary coal or coke requires for its complete combustion 12 lbs. of air per pound of the fuel, but in practice, with chimney draught, twice this quantity may be supplied, as much passes free through the furnace in an uncombined state. Kennedy found that 18 lbs. of air gave the best results.

At the temperature of 62° F. and at the sea-level, the volume of 1 lb. of air is 13·14 cubic feet; therefore 315 cubic feet of air are necessary for the proper combustion of 1 lb. of coal or coke in an ordinary furnace. If artificial or forced draught is employed, this quantity may be reduced to about 250 cubic feet, more or less, according to the force of the draught.

The following Table gives additional information as to the composition, total heat of combustion, and evaporative value of various fuels:—

* Allowing for the 212° below which there can be no abstraction of heat for evaporation, the multiplier should be 14·78 instead of 15.

Table LXXXVII.—Composition and Value of Fuels.

Description.	Carbon.	Hydrogen.	Oxygen.	Sulphur.	Ash, &c. including Nitrogen.	Total Heat of Combustion.	Evaporation from and at 212° or B.T.U.÷966°	One ton occupies in Cubic Feet.
						Units.	Pounds.	
Welsh—Ebbw Vale,	87·78	5·15	0·39	1·02	3·66	16221	16·79	...
,, Powell's Duffryn,	88·26	4·66	0·60	1·77	4·71	15788	16·34	...
,, Llangennech,	84·97	4·26	3·50	0·42	6·85	14682	15·20	...
,, Graigola,	84·87	3·84	7·19	0·45	1·91	14130	14·63	...
,, Average,	83·87	4·79	4·15	1·43	5·89	14858	15·52	42·7
Newcastle Average,	82·12	5·31	5·69	1·24	5·12	14820	15·32	45·3
Derbyshire Average,	79·68	4·94	10·28	1·01	4·06	13860	14·34	47·4
South Yorkshire,	81·88	4·83	7·47	0·54	2·95	14296	14·71	46·0
Lancashire Average,	77·90	5·32	9·53	1·44	6·18	13918	14·56	45·2
Scotch ,,	78·53	5·61	9·69	1·11	5·03	14164	14·65	42·0
Irish Anthracite,	80·03	2·30	...	6·76	11·03	13302	14·5	35·7
American Anthracite,	88·54	2·43	...	0·04	8·60	15163	15·67	42·35
,, Bituminous,	73·21	...	...	0·42	11·27	...	...	42·44
French Anthracite,	86·17	2·67	2·85	...	8·56	14038	14·53	40·00
,, Hard Bituminous,	88·56	4·88	4·38	...	2·19	15525	16·10	42·75
,, Caking ,,	87·73	5·08	5·65	...	1·54	15422	16·00	42·75
Chilian Coal,	63·56	5·43	14·84	2·50	14·13	11030	11·68	...
Indian ,, Average,	70·20	...	...	...	22·9	...	...	...
Patent Fuel—Warlichs,	90·02	5·56	...	1·62	...	16495	17·07	...
,, ,, Average,	83·40	4·97	2·79	1·26	6·01	15000	15·66	34·4
Natal Coal—Navigation dry	81·38	5·45	2·82	1·06	9·29	14045	14·54	...
Kentish Coal	79·78	...	16·73	0·8	2·08	13900	14·4	...
Coke—Best Durham,	85 to 92	...	...	0·25 to 2·0	4 to 12	12832	13·30	...
Woods—Oak,	49·64	5·92	41·16	...	3·26	average		94·4
,, Birch,	50·20	6·20	41·62	...	1·96	when dry		106·1
,, Fir,	51·79	6·28	41·93	...	...	7800	8·1	...
Peat—Fairly dry,	59·6	5·80	29·6	0·3	4·7	9951	10·30	...
Petrol'm—Americ'n crude	84·7	13·1	2·2	...	...	20240	20·33	39 to 46
,, Refuse,	87·1	11·7	1·2	...	...	19832	20·53	40
,, Caucasian crude,	86·6	12·3	1·1	...	...	20138	20·85	42

Table LXXXVIIa.—Liquid Fuels. Composition and Heating Value of Oils (*from Redwood*).

	Specific Gravity at 32° F.	Chemical Composition.			Actual evaporation per lb. of Fuel from 212° F.	British Thermal Units per lb.
		Carbon.	Hydrogen.	Oxygen.		
		Per cent.	Per cent.	Per cent.	Lbs.	
Heavy petroleum from West Virginia	0·873	83·5	13·3	3·2	14·58	18,324
Light petroleum from West Virginia	0·841	84·3	14·1	1·6	14·55	18,401
Light petroleum from Pennsylvania	0·816	82·0	14·8	3·2	14·05	17,938
Heavy petroleum from Pennsylvania	0·886	84·9	13·7	1·0	15·30	19,210
American petroleum	0·820	83·4	14·7	1·9	14·14	17,588
Petroleum from Parma	0·786	84·0	13·4	1·8	13·96	18,218
,, Pechelbronn	0·912	86·9	11·8	1·3	14·30	17,474
,, ,,	0·892	85·7	12·0	2·3	14·48	18,036
,, Schwabweiler	0·861	86·2	13·3	0·5	15·36	18,824
,, East Galicia	0·870	82·2	12·1	5·7	14·23	18,153
,, West Galicia	0·885	85·3	12·6	2·1 (NO)	14·79	18,416
Shale oil from Ardeche, France	0·911	80·3	11·5	8·2 (0·5 N)	12·24	16,283
Coal tar from Paris Gas Works	1·044	82·0	7·6	10·4	12·77	16,049
Petroleum from Balakhany	0·882	87·4	12·5	0·1	17·41	21,060
Light petroleum from Baku	0·884	86·3	13·6	0·1	16·40	20,628
Heavy ,, ,,	0·938	86·6	12·3	1·1	15·55	19,440
Petroleum residues from the Baku factories	0·928	87·1	11·7	1·2	15·54	19,260
Petroleum from Java	0·923	87·1	12·0	0·9	15·02	19,496
Heavy oil from Ogaio	0·985	87·1	10·4	2·5	14·75	18,146
,, ,, ,, Mexico, crude	0·950	83·5	11·7	1·37 (NO)	...	18,900
,, ,, ,, California, crude	0·966	81·5	11·6	6·92 (NO)	...	18,670
,, ,, ,, Texas, crude	0·924	84·6	10·9	2·57	...	18,800

Table LXXXVIIb.—Consumption of Various Oil Fuels in Gallons per Twenty-four Hours.

N.B.—For Scotch Shale, multiply by 0·948; for Good Crude, Specific Gravity (0·9), multiply by 0·905.

B.H.P.	Consumption of Paraffin, Specific Gravity 0·815, in fractions of lbs. per B.H.P. hour.													
	0·300	0·325	0·350	0·375	0·400	0·425	0·450	0·475	0·500	0·525	0·550	0·575	0·600	0·650
50	44·18	47·86	51·54	55·22	58·90	62·58	66·26	69·94	73·62	77·30	80·98	84·66	88·34	95·7
75	66·26	71·78	77·30	82·83	88·35	93·87	99·39	104·9	110·4	116·0	121·5	127·0	132·5	143·6
100	88·35	95·71	103·6	110·4	117·8	125·2	132·5	139·9	147·2	154·6	162·0	169·3	176·7	191·4
125	110·4	119·6	128·8	138·1	147·3	156·5	165·7	174·9	184·1	193·3	202·5	211·7	220·9	239·3
150	132·5	143·5	154·6	165·7	176·7	187·7	198·8	209·8	220·9	231·9	243·0	253·1	265·0	287·1
175	154·6	167·5	180·4	193·3	206·2	219·0	231·9	244·8	257·7	270·6	283·4	296·3	309·2	335·0
200	176·7	191·4	206·2	220·9	235·6	250·3	265·0	278·8	294·5	309·2	323·9	338·6	353·4	382·8
225	198·8	215·4	232·0	248·5	265·1	281·6	298·2	314·7	331·3	347·9	364·4	381·0	397·5	430·7
250	220·9	239·3	257·7	276·1	294·5	312·9	331·3	349·7	368·1	386·5	404·9	423·3	441·7	478·5
275	243·0	263·3	283·5	303·7	324·0	344·2	364·4	384·7	404·9	425·2	445·4	465·3	485·9	526·4
300	265·1	287·2	309·3	331·3	353·4	375·5	397·6	419·7	441·7	463·8	485·9	507·7	529·9	574·2
325	287·1	311·0	334·9	358·9	382·9	406·8	430·7	454·6	478·5	502·5	526·4	550·0	573·2	622·1
350	309·2	335·0	360·7	386·5	412·3	438·1	463·8	489·6	515·3	541·1	566·9	592·3	618·4	669·9
400	353·4	382·9	412·3	441·7	471·2	500·6	530·1	559·5	588·9	618·4	647·8	677·0	706·7	765·6
450	397·6	430·7	463·9	496·9	530·1	563·2	596·3	629·5	662·6	695·7	728·8	761·6	795·1	861·3
500	441·8	478·6	515·4	552·2	589·0	625·8	662·6	699·4	736·2	773·0	809·8	846·6	883·4	957·0
550	485·9	526·4	566·9	607·4	647·9	688·4	728·9	769·4	809·8	850·3	890·8	931·3	971·2	1053
600	530·1	574·2	618·4	662·3	706·8	751·0	795·1	839·3	883·4	927·6	971·8	1016	1060	1147
650	574·3	622·2	670·0	717·8	765·7	813·5	861·4	909·2	957·1	1005	1053	1101	1148	1243
700	618·5	670·1	721·6	773·0	824·6	876·1	927·6	979·2	1031	1082	1134	1185	1237	1339
750	662·6	717·8	773·0	828·3	883·5	938·7	993·9	1049	1104	1160	1215	1270	1325	1435
800	706·8	765·7	824·6	883·5	942·4	1001	1060	1119	1178	1237	1296	1355	1413	1530
850	751·0	813·6	876·2	938·7	1001	1064	1126	1189	1252	1314	1377	1439	1502	1626
900	795·1	861·4	927·5	993·9	1060	1126	1192	1259	1325	1391	1458	1524	1590	1722
950	839·3	909·3	979·0	1049	1119	1189	1258	1329	1399	1468	1530	1609	1678	1818
1000	883·5	957·1	1031	1104	1178	1251	1325	1399	1472	1546	1620	1693	1767	1913

Table LXXXVIIc.—Liquid Fuels as in Practice.

Fuel.	Specific Gravity.	Ash, per Cent.	Water, per Cent.	Sulphur, per Cent.	Gross Heat Value.		Nett Heat Value.	
					Per Lb.	Per Gallon.	Per Lb.	Per Gallon.
					B.T.U.	B.T.U.	B.T.U.	B.T.U.
British Navy oil . .	0·907	0·023	0·500	0·750 *	19,250	174,598	17,900	162,353
Roumanian gas oil . .	0·871	...	trace	...	19,280	167,929	18,010	156,867
,, ordinary oil .	0·928	0·051	,,	0·430	18,920	175,578	17,685	164,117
,, residues . .	0·930	0·133	,,	...	19,010	176,793	17,730	163,889
Trinidad crude oil . .	0·964	0·141	,,	3·360	18,400	177,376	17,230	166,097
Tarakan ,, . .	0·948	0·035	,,	0·250	18,870	178,822	17,650	161,322
Californian ,, . .	0·945	...	...	...	18,080	170,856	16,900	159,705
Mexican ,, . .	0·950	...	...	3·270	18,800	178,600	17,580	167,010
Texas ,, . .	0·924	...	...	1·63	19,000	175,560	17,770	164,195
Texas and Californian residuals	...	...	...	0·90	19,200	...	18,000	...

* The Admiralty now permit up to 3 per cent of Sulphur in Oil Fuel for steam boilers, *v.* p. 308.

Table LXXXVIId.—Oil Fuels as Used in U.S. American Navy Boilers.

	Crude Texas Oil.	Texas Oil as Used.	Californian Oil as Used.	Residuum of Mixed Texas and Californian.	Coal, Pocahontas Handpicked and Screened.
Carbon, content per cent.	84·60	83·26	81·52	84·35	85·94
Hydrogen ,, ,,	10·90	12·41	11·01	11·33	4·45
Oxygen ,, ,,	2·87	3·83	6·92	2·82	4·50
Nitrogen ,, ,,	...	...		0·60	1·14
Sulphur ,, ,,	1·63	0·50	0·55	0·90	0·82
Specific gravity at 60° F.	0·924	0·926	0·966	0·966	...
Flash point . . F.°	180	216	311	270	...
Fire ,, . . ,,	200	240	...	280	...
Vaporisation point ,,	...	142	230	...	...
Loss for 6 hours at 212° F.	...	21·65%	12·20%	14·90%	...
Calorific value as calculated	18,812	19,480	18,192	19,049	14,992
Calorific value by calorimeter . . .	19,060	...	18,667	19,215	...
Water evaporated from and at 212° F. in a Hohenstein w.t. boiler	...	12·654	11·574	11·515	9·215
Water evaporated from and at 212° F. in a Hohenstein w.t. boiler best results with air pulverisers . . .	...	12·850	...	12·660	...

Table LXXXVIIe.—Liquid Fuels, Viscosity of, at various Temperatures, by Number of Seconds taken for the Passing of 12·2 cubic inches (200 c.cm.).

Fuel.	Specific Gravity.	Temperature, F.					
		70°.	85°.	105°.	120°.	140°.	160°.
British Navy oil .	0·907	818	435	255	179	140	110
Roumanian residues	0·946	...	...	...	785	474	293
Trinidad crude oil .	0·964	...	...	...	...	...	1127
Tarakan ,, .	0·948	745	431	263	170	127	99
Mexican ,, .	0·950	...	...	...	...	...	...

Rules for the Burning and Carrying of Oil Fuel.

1. In vessels fitted for burning oil fuel, the following records will be made in the Register book:—"Fitted for oil fuel F.P. above 150° F." in cases in which approval has been given for the use of high flash point oil only; and "Fitted for low flash oil fuel" in cases in which the approval covers the use of low flash point.

2. The following arrangements are applicable only to the case of oil fuel, the flash point of which (by Abel's close test) is not below 150° F. For a lower flash point the arrangements must be submitted for consideration.

3. Oil fuel, the flash point of which does not fall below 150° F., may be carried in ordinary cellular double bottoms under engines or boilers or ordinary cargo holds, in peak or in deep tanks, or in oil bunkers specially constructed.

4. Cellular double bottoms when fitted for oil fuel are to have oil-tight centre line divisions, and the lengths of these compartments are to be submitted for approval.

5. Peak tanks, deep tanks, bunkers specially constructed for oil fuel, and settling and other service tanks must be fitted with bulkhead subdivisions or wash plates to the Committee's satisfaction and strengthened to efficiently withstand the stresses on them when only partly filled and in a seaway. The riveting of these spaces is to be as required by the Rules in the cases of vessels carrying petroleum in bulk, and the scantlings and arrangements to the Committee's satisfaction.

6. All compartments intended for carrying oil fuel must be tested by a head of water extending to the highest point of the filling pipes, or 12 feet above the load-line or 12 feet above the highest point of the compartment, whichever is the greatest.

7. Each compartment must be fitted with an air-pipe, to be always open, discharging above the upper deck. All double-bottom compartments used for oil fuel should have suitable holes and doors of approved design fitted in the outer bottom plating.

8. The pumping arrangements of the oil-fuel compartments must be absolutely distinct from those of other parts of the vessel and submitted for approval. All oil-fuel suction pipes and all bilge or ballast pipes passing through oil-fuel tanks or oil bunkers are to be of iron or steel.

9. If it is intended to carry sometimes oil-fuel and sometimes water ballast in any of the compartments, the valves or cocks connecting the suction pipes to these compartments with the ballast donkey pump and those controlling them with the oil-fuel pump must be so arranged that oil may be pumped from any one compartment by the oil-fuel pump at the same time as the ballast donkey is being used on any other compartment.

10. All oil-fuel suction pipes should have valves or cocks fitted at the bulkheads where they enter the machinery space, capable of being worked both from this space and from the deck, outside of the fidely and engine-room casings. Valves or cocks similarly worked are to be fitted to all pipes leading from the settling or other tanks.

11. Oil-fuel pipes should, where practicable, be placed above the stokehold and engine-room plates, and where they are always visible.

12. No wood fittings or bearers are to be fitted in the stokehold spaces.

13. Where oil-fuel compartments are at the sides of, or above, or below the boilers, special insulation is to be fitted where necessary to protect them from the heat from the boilers, smoke boxes, casings, &c.

14. Water service pipes and hoses are to be fitted so that the stokehold plates can at any time be flushed with sea water into the bilges.

15. If the oil fuel is sprayed by steam, means are to be provided to make up for the fresh water used for this purpose.

16. If the oil fuel is heated by a steam coil, the condensed water should not be taken directly to the condensers, but should be led into a tank or an open funnel mouth, and thence led to the hot well or feed tank.

Admiralty Conditions of Contract for Oil Fuel.

1. *Quality.*—The oil fuel supplied under this contract shall consist of liquid hydrocarbons, and may be either:—

(*a*) Shale oil; or

(*b*) Petroleum as may be required; or

(*c*) A distillate or a residual product of petroleum;

and shall comply with the Admiralty requirements as regards flash point, fluidity at low temperatures, percentage of sulphur, presence of water, acidity, and freedom from impurities.

The flash point shall not be lower than 175° F., close test (Abel or Pensky-Martens).*

The proportion of sulphur contained in the oil shall not exceed 3 per cent.

The oil fuel supplied shall be as free as possible from acid, and in any case the quantity of acid must not exceed 0·5 per cent., calculated as oleic acid when tested by shaking up the oil with distilled water, and determining by titration with decinormal alkali the amount of acid extracted by the water, methyl orange being used as indicator.

The quantity of water delivered with the oil shall not exceed 0·5 per cent.

The viscosity of the oil supplied shall not exceed 2000 seconds for an outflow of 50 cubic centimetres at a temperature of 32° F., as determined by Sir Boverton Redwood's standard viscometer (Admiralty type for testing oil fuel).

The oil supplied shall be free from earthy, carbonaceous, or fibrous matter, or other impurities which are likely to choke the burners.

The oil shall, if required by the inspecting officer, be strained by being pumped on discharge from the tanks, or tank steamer, through filters of wire gauze having 16 meshes to the inch.

The quality and kind of oil supplied shall be fully described. The original source from which the oil has been obtained shall be stated in detail, as well as the treatment to which it has been subjected, and the place at which it has been treated.

The ratio which the oil supplied bears to the original crude oil should also be stated as a percentage.

Rate of combustion.—In the mercantile marine, when working economically with chimney draught only, the coal burned per square foot of grate per hour varies from about 15 lbs., with long bars (6 feet to 6 feet 6 inches), to about 20 lbs. when the length of the bars does not exceed 1·33 × diameter of furnace, and all calculations for grate areas of merchant steamers should be based on these figures; for, although larger quantities may be *consumed* by forcing the fires, or may, perhaps, be completely burned when the wind is strong and draught good, there will be, on the other hand, many days when the quantity burned will be less than the average given above.

* In the case of oils of exceptionally low viscosity, such as distillates from shale, the flash point must be not less than 200° F.

With forced or artificial draught much larger quantities are consumed: with the Howden system as much as 40 lbs. can be economically burnt, and even up to 60 lbs. with somewhat less economic results.

The higher efficiency of the shorter bars is largely due to the better air supply, since, with a given diameter of furnace, the area at the mouth of the ash-pit is the same whether the bars are long or short; and it is no doubt for this reason that the Macfarlane Gray rule—that *the consumption of coal per foot of grate is very nearly proportional to the diameter of the furnace*—is found to be so nearly correct in everyday practice. Another cause of the higher efficiency of short bars is the fact that bars over about 5 feet in length cannot be so well stoked by hand, with the result that the fire burns into holes at the back end, and allows cold air to rush in,—thus depriving the other parts of the fire of their due proportion of air and reducing the efficiency of the furnace. In practice, with average stokers, the efficiency of a grate over 4 feet in length is nearly inversely as its length.

The rate of combustion also naturally depends a good deal on the quality or class of coal burned, but by far the most important factor is the strength of the draught.

Chimney or funnel.—The draught obtained with any chimney depends mainly on its sectional area, its height, and the difference of absolute temperature between the gases in it and the external air.

Professor Rankine gave the following formulæ for chimney draught:—

Let w be the weight of fuel burned in a given furnace per second in pounds.

V_0 the volume at 32° of the air supplied per pound of fuel.

τ_0 the *absolute* temperature at 32° Fahr., which is 461° + 32°.

τ_1 the absolute temperature of the gas discharged by the chimney, whose sectional area is A: then,—

Rule 241. Velocity of the current in the chimney in feet per second is

$$= \frac{w \times V_0 \times \tau_1}{A \times \tau_0}.$$

The density of that current in pounds to the cubic foot is very nearly

$$= \frac{\tau_0}{\tau_1}\left(0{\cdot}0807 + \frac{1}{V_0}\right);$$

that is to say, from 0·084 to 0·987 × ($\tau_0 \div \tau_1$).

Let l denote the whole length of the chimney, and of the flue leading to it, in feet;

m its "hydraulic mean depth"; that is, its area divided by its perimeter; which, for a square or round flue or chimney, is one quarter of the diameter or side;

f, a co-efficient of friction, whose value for currents of gas moving over sooty surfaces is estimated by Peclet at 0·012;

G, a factor of resistance for a passage of air through the grate and the layer of fuel above it, whose value, according to the experiments of Peclet on furnaces burning from 20 to 24 pounds of coal per square foot of grate per hour, is 12.

Then, according to Peclet's formula,—

Rule 242. The *head* required to produce the draught in question is

$$=\frac{\mu^2}{2g}\left(1+G+\frac{f.l}{m}\right),$$

which, with the values assigned by Peclet to the constants, becomes

$$=\frac{\mu^2}{2g}\left(13+\frac{0{\cdot}012\times l}{m}\right).$$

When the *head* is given the value of μ may be calculated, and then,

Rule 243. Weight of fuel which the furnace is capable of burning *completely* per hour

$$=\frac{\mu\times A\times\tau_0}{V_0\times\tau_1}.$$

It is usual to reckon the *head* by taking one inch of water as the unit; then,—

$$\text{Head in inches of water}=0{\cdot}192\times h\times\frac{\tau_0}{\tau_1}\left(0{\cdot}0807+\frac{1}{V_0}\right),$$

Sir J. Thornycroft found, by careful experiment with loco. boilers in torpedo-boats working with a *plenum* (that is, with a closed stoke-hole into which air is forced), "that of the initial pressure, the resistance of the tubes accounts for about seven-tenths of the whole, the resistance of the fire and fire-bars being only about one-tenth;" and that "the pressure in the funnel, as measured, was sensibly equal to atmospheric pressure."

Professor Rankine also stated that if H be the height of the funnel, τ_2 the absolute temperature of the external air, then,—

Rule 244. Head produced by chimney draught $=H\left(0{\cdot}96\frac{\tau_1}{\tau_2}-1\right)$,

or, taking h as the head,

Rule 244a. Height of chimney required to produce a given draught

$$=h\div\left(0{\cdot}96\frac{\tau_1}{\tau_2}-1\right).$$

The velocity of the gas in the chimney is proportional to $\sqrt{h}$, and therefore to $\sqrt{0{\cdot}96\tau_1-\tau_2}$.

The density of that gas is proportional to $\frac{1}{\tau_1}$.

The weight discharged per second is proportional to velocity × density, and therefore, to $\frac{\sqrt{0{\cdot}96\tau_1 - \tau_2}}{\tau_1}$; which expression becomes a maximum, when $\tau_1 = \frac{25}{12}\tau_2$. Therefore the best chimney draught takes place when the absolute temperature of the gas in the chimney is to that of the external air as 25 to 12.

When this condition is fulfilled $h = H$.

That is, the height of the chimney for the best draught is equal to the *head* expressed in hot gas, and the density of the hot gas is half that of the air.

From the above it appears that the best temperature of gases at the base of the funnel is about 600°, and that therefore nearly one-fourth of the total heat of combustion is absorbed in creating the draught. The temperature of steam at 250 lbs. pressure absolute is 400° F.; it is therefore necessary, in order to maintain the efficiency of the heating surface, to have a temperature at funnel base in excess of this, and when there is a superheater requiring a temperature of 550° to 600° F., it is obvious that the hot gases when leaving it will be above those, so that with modern conditions 600° at the funnel base is really low, and it is not uncommon to have 700° to 750° there with the proportions of heating surface to grate surface, size of tubes, &c., commonly employed; but lower temperatures can easily be reached by using artificial draught and small tubes (say 2½ inch) fitted with retarders.

The diameter and height of funnel are often fixed by arbitrary methods rather than rules. The following, however, are simple and easily handled.

Rule 244b. Height of chimney in feet $= \cdot 007\left(\frac{C}{A}\right)^2$.

Rule 244c. Area of chimney in square feet $= \frac{C \times \cdot 084}{\sqrt{H}}$.

Where H = Height in feet.
A = Area of section in square feet.
C = Consumption in lbs. per hour on grates.

The following is a rough rule for modern practice:—

Rule 244d. Diameter of chimney in inches $= \sqrt{I.H.P \times F}$.

For Ordinary Merchant Steamers	F = 2·38 Natural Draft.
,, Naval and Express Steamers	F = 2·16 Assisted ,,
,, ,, ,, ,,	F = 1·66 Forced ,,

When the draught depends on the chimney alone, 40 feet should be regarded as the minimum height, where possible; as the size of the vessel increases it is, of course, possible to get longer chimneys, and, in consequence, greater draught pressures, without fans. In the "Olympic" the tops of the chimneys are at 160 feet above the dead plates.

Table LXXXVIII.—Capacity of Funnels for quantities of Fuel burnt in lbs. per hour.

Diameter of Funnel.	Height of Funnel above dead plates in feet.														
	20	25	30	35	40	50	60	70	80	90	100	110	120	130	140
Feet.															
2·00	167	187	207	224	239	264	288	311	333	..	..	..	..	..	..
2·25	213	239	261	282	303	337	370	399	426	..	..	..	..	..	..
2·50	261	292	320	346	370	413	453	489	522	..	..	..	..	..	..
2·75	316	354	387	419	448	500	548	591	632	..	..	..	..	..	..
3·00	..	421	464	500	535	595	653	705	753	799	..	..	..	..	..
3·25	..	495	543	586	627	700	767	829	885	940	..	..	..	..	..
3·50	..	575	630	681	728	813	891	963	1,028	1,091	..	..	..	..	..
3·75	..	660	723	780	834	932	1022	1,103	1,180	1,251	..	..	..	..	..
4·00	..	750	822	888	948	1061	1161	1,254	1,341	1,422	1,496	..	..	..	..
4·25	..	..	926	1000	1070	1195	1310	1,415	1,510	1,604	1,692	..	..	..	..
4·50	..	..	1040	1123	1291	1342	1471	1,589	1,697	1,801	1,898	..	..	..	..
4·75	..	..	1156	1250	1337	1492	1636	1,767	1,888	2,004	2,113	..	..	..	..
5·00	..	..	1282	1385	1480	1654	1812	1,956	2,088	2,221	2,340	2,455	..	..	..
5·25	..	..	..	1527	1633	1824	2000	2,159	2,307	2,448	2,580	2,706	..	..	..
5·50	..	..	..	1676	1792	2000	2192	2,364	2,528	2,733	2,880	3,026	..	..	..
5·75	..	..	..	1829	1956	2185	2395	2,586	2,762	2,932	3,085	3,240	..	..	..
6·0	..	..	..	..	2140	2380	2609	2,820	3,012	3,196	3,366	3,531	3,687	3,842	..
6·5	..	..	..	..	2508	2800	3068	3,316	3,540	3,760	3,955	4,154	4,336	4,514	..
7·0	..	..	..	..	2912	3252	3564	3,852	4,112	4,364	4,580	4,804	5,015	5,221	5,438
7·5	..	..	..	..	..	3728	4088	4,412	4,720	5,004	5,264	5,518	5,760	5,996	6,223
8·0	..	..	..	..	..	4228	4644	5,016	5,448	5,688	5,980	6,283	6,559	6,818	7,074
8·5	..	..	..	..	..	4780	5240	5,660	6,040	6,416	6,768	7,100	7,413	7,756	8,009
9·0	..	..	..	..	..	..	5884	6,356	6,788	7,204	7,592	7,961	8,311	8,653	8,979
9·5	..	..	..	..	..	..	6544	7,068	7,552	8,016	8,452	8,864	9,253	9,633	9,996
10·0	..	..	..	..	..	..	7248	7,824	8,352	8,885	9,360	9,820	10,249	10,670	11,078
11·0	..	..	..	..	..	..	..	9,466	10,111	19,738	11,314	11,844	12,384	12,863	13,379
12·0	..	..	..	..	..	..	..	11,280	12,050	12,784	13,464	14,124	14,748	15,368	15,925

Ordinarily, in the merchant service, the area of chimney is about a fifth to a quarter of the grate area, whilst in the Navy they are seldom larger that 0·14, and vary from that to 0·11, according to the air-pressure intended to be used.

The thickness of funnel plates may be approximately as follows:—

Upper plates	...	·1-inch + (·01 inch for each foot of diameter).
Middle ,,	...	·125 ,, + (·01 ,, ,,).
Lower ,,	...	·15 ,, + (·01 ,, ,,).

Funnels should be suitably stiffened by angle, tee, or channel rings or hoops, and large ones cross-stayed and stiffened longitudinally.

The funnels of Naval ships are made considerably lighter, the upper plates being usually 6¾ lbs. per sq. foot, even for the largest sizes.

Table LXXXVIIIa.—Pitch, &c., of Riveting for Funnels, Casings, &c. (Admiralty work).

Description.	Thickness of Plate.	Diameter of Rivet.	Pitch of Rivet.
	Inch.	Inch.	Inches.
Uptakes,	$\frac{3}{16}$	$\frac{5}{8}$	2
	$\frac{1}{4}$	$\frac{1}{2}$	2
	$\frac{5}{16}$	$\frac{5}{8}$	$2\frac{1}{2}$
	$\frac{3}{8}$	$\frac{5}{8}$	$2\frac{1}{2}$
Funnels,	$\frac{1}{4}$	$\frac{1}{2}$	3
	$\frac{5}{16}$	$\frac{5}{8}$	2
	$\frac{1}{8}$	$\frac{5}{8}$	2
	$\frac{1}{16}$	$\frac{1}{4}$	2
Funnel casing,	$\frac{3}{16}$	$\frac{5}{8}$	3
	$\frac{1}{8}$	$\frac{5}{8}$	2
	$\frac{1}{16}$	$\frac{1}{4}$	2
Deck casing,	$\frac{3}{16}$	$\frac{5}{8}$	2
	$\frac{1}{8}$	$\frac{5}{8}$	2
	$\frac{1}{16}$	$\frac{1}{4}$	2
Cowls and trunks (exposed to weather),	$\frac{1}{8}$	$\frac{5}{8}$	2
	$\frac{1}{16}$	$\frac{1}{4}$	2
Screens and ventilating trunks,	$\frac{1}{8}$	$\frac{5}{8}$	4
	$\frac{1}{16}$	$\frac{1}{4}$	3
Other work,	$\frac{3}{16}$	$\frac{5}{8}$	2 to 3
	$\frac{1}{8}$	$\frac{5}{8}$	2 to 3
	$\frac{1}{16}$	$\frac{1}{4}$	2

When a number of boilers are served by the same funnel, and some of the furnaces are much further away from the base of the funnel than others (as is sometimes unavoidably the case in Naval vessels) dampers for regulating the force of the draught should be fitted in each "leg" of the uptakes.

Similarly,—proper arrangements (flues, screens, &c.) should be fitted to distribute the air in the stoke-holes, when the "closed stoke-hole" system of forced draught is used.

Smoke-box doors should be made "three thick,"—and arrangements provided to obtain a good current of air between the inner and central plates, whilst the space between the central and outer plates should be filled with some good non-conducting substance.

Forced draught.—In the Navy the "closed stoke-hole" system of forced draught is now generally used; as the name implies, the whole stoke-hole is made air-tight, and the fans force air down into it,—the men working under pressure and entering and leaving through air locks.

Of the "closed ash-pit" systems, the Howden is the one now generally adopted in the mercantile marine. In this system air is led by pipes from the fans to each furnace separately, and can thus be shut off from any one before opening the door, so that the rush of cold air, which is such an objectionable feature in the closed stoke-hole system, is entirely avoided. The Howden also heats the air supply to 180° to 200° by passing it amongst a number of tubes placed in the uptake, and thus no doubt materially increases the efficiency of the furnace, since this increase of temperature so facilitates combustion as to give a gain much in excess of the actual amount of heat stored in the air.

Messrs Howden state that the horse-power per square foot of grate obtained from boilers fitted on this system varies from about 15 on tramp steamers working with very poor bunker coal, to over 26 on cruisers using best Welsh steam coal; and that the corresponding heating surfaces per I.H.P. would be about 2·8 square feet and 1·6 square feet.

As regards coal consumption, Messrs Howden consider that—in the case of triple engines working with 160 lbs. to 180 lbs. steam pressure —1·2 lbs. of Cardiff or equally good coal per I.H.P. per hour should not be exceeded in regular work at sea.

The air pressure used in the ash-pits varies from ½ inch to 1½ inch, according to the circumstances of the case.

An air temperature of about 300° is now aimed at, and the effect of the abstraction of this heat from the waste gases is to materially lower their temperature. The air tubes are, as a rule, 2¾ inches external diameter and 14 L.S.G. in thickness. Their length varies from 30 inches to 60 inches.

Coal consumptions up to 50 lbs. per square foot of grate per hour have been effected without the slightest trouble.

Mr Howden claims for his system the greatly reduced space and weight which must be allotted to boilers, the reduced coal consump-

tion, and the very uniform temperature of furnaces, all of which latter tend to reduce repairs and to lengthen the life of the boiler.

The great advantage of artificial or forced draught is that at all times it is completely under control, and quite independent of wind or weather, thus ensuring a uniform efficiency of furnace with all ordinary qualities of coal, and any rate of combustion that may be required. In Naval ships this is a necessity,—since the chimney may be entirely destroyed at any moment.

Pressures of air in stokeholds, &c., are usually reckoned in inches of water (often called for shortness "inches" simply), and are measured by a "water-gauge," **U** form, partly filled with water, and having the open end of one leg exposed to the pressure to be measured, and the other to the outer air. The pressure is indicated by the difference in level of the water in the two legs.

Stokehold ventilators.—The downcast pipes with natural draught should have an aggregate transverse area of 0·45 square inch per lb. of fuel burnt per hour; or 0·675 square inch per I.H.P. in cargo boats, 0·75 in express steamers; 0·62 is sufficient for turbine-driven ships.

The area of the cowl mouth should be not less than

1·35	square inch	per lb.	fuel per hour	in 10	knot ships.
1·24	,,	,,	,,	12½	,,
1·13	,,	,,	,,	15	,,
0·93	,,	,,	,,	20	,,
0·78	,,	,,	,,	25	,,

Table LXXXIX. shows the results of tests of the various modern boilers with oil and coal fuel.

Table XC. shows the rates of combustion and of evaporation usual on sea trials of various classes of vessel under the conditions named.

With inferior coal, supplied at foreign stations, the figures in columns 5 and 6 may be reduced as much as 20 per cent., whilst with picked coal and highly skilled stokers, they may be increased nearly 10 per cent.

The feed temperatures usual in the respective classes of vessels are assumed, and allowance should be made for any special feed-heating.

The evaporative efficiency of the boilers of cruisers and battleships appears lower than it really is, as a large amount of steam is consumed by the numerous auxiliary engines and for domestic purposes, but is left out of consideration here; for quantities of steam so consumed see Tables XCII. and XCIII.

By way of a comparison with Table XC., it may be noted that a goods locomotive consumes about 40 lbs. of coal per foot of fire-grate per hour, and an express passenger engine from 65 to 80 lbs.; and, as the evaporative efficiency of the locomotive boiler is high, the water evaporated per square foot of grate per hour may be taken as about 420 lbs. for the goods engine, and from 650 to 750 lbs. for the passenger engine.

Table LXXXIX.—Results of Trials of

	U.S.A. Hohenstein W.T. Boiler.			B.	C. 12·5 diam., &c.			Yarrow Naval, Light Weight.
	Natural Draught.	Fcd. Dght. = 2·1 ins.	Fcd. Dght. = 1·0 in.	12 ft. diam., 11 ft. long. 2 Furnaces.	Nat. Dght.	Nat. Dght.	Fcd. Dght. = 1·5 ins.	
Type of boiler . .	Water tube	Water tube	Water tube	Cylindrical	Cylindrical	Cylindrical	Cylindrical	Water tube
Total heating surface, sq. ft.	2,130	2,130	2,130	1,200	1,695	1,695	1,695	8,300
Grate area, sq. ft. .	50·14	50·14	50·14	..	40	40	40	..
Ratio of T.H.S. to grate area.	42·5	42·5	42·5	..	42·3	42·3	42·3	..
Working pressure, lbs.	275	275	275	140	120	120	120	250
Fuel, kind of . .	Pocahontas coal	Coal	Texas oil	Texas oil	Newcastle coal	Texas oil	Texas oil	Texas oil
Fuel value in B.T.U.	14,992	14,992	19,481	19,000	14,500	19,000	19,000	19,000
Engines supplied .	..	..	..	..	..	..	..	Turbines
Engines, horse-power, per boiler.	..	..	..	..	..	..	..	8,900
Fuel consumed per hr., lbs.	1,037	2,674	1,969	820	974	633	1,222	9,800
Fuel consumed per sq ft. of grate.	20·68	50·33	..	..	24·35	..	..	..
Fuel consumed per sq. ft. of T.H.S.	0·49	1·26	0·92	0·68	0·58	0·37	0·72	1·12
Fuel consumed per H.P.	..	..	..	..	..	..	..	1·033
Water evaporated per hour, lbs.	8,769	21,549	21,309	10,917	7,558	7,756	14,591	107,000
Water evaporated per lb. of fuel.	8·46	8·06	10·82	13·31	7·76	12·25	..	11·70
Water evaporated per lb. of fuel from and at 212°.	9·645	9·38	12·37	16·13	9·31	14·45	14 06	13·68
Water evaporated per sq. ft. of T.H.S.	4·12	10·12	10·00	9·09	4·46	4·58	8·61	12·84
Water evaporated per H.P.	..	..	..	..	..	..	..	12·05
Total heating surface per H.P.	..	..	..	..	..	..	..	0·93
Efficiency per cent. .	62·11	60·41	61·34	82·08	62·00	73·46	71·48	71·47
Total weight of boiler, &c., per ft. T.H.S., lbs.	25·4	25·4	..	..	..	..	..	13·34
Total weight of boiler, &c., per H.P., lbs.	..	..	..	..	..	..	..	12·45
Total weight of boiler, &c., lbs. water evaporated per hour.	6·16	2·51	2·51	..	..	..	..	1·035

Boilers at Full Power.

Thornycroft, Light Naval, Fcd. Dght.	White-Forster Naval Boilers, Light Naval.			Babcock-Wilcox, U.S.A. Naval Boiler.		Miyabara, Japanese Naval Boiler.		Ordinary Mercantile. 16·2 ft. diam., 11·5 ft. long.	Niclausse Naval, Fcd. Dght.	Belleville Naval, Fcd. Dght.
	Fcd. Dght.	Nat. Dght.	Fcd. Dght.	Nat. Dght.	Fcd. Dght.	Fcd. Dght.	Fcd. Dght.			
Water tube	Water tube	Water tube		Water tube		Water tube		Cylindrical	Water tube	Water tube
7,930	7,500	4,000		5,353		1,912		2,994	8,000	4,099
..	125·0	..		119		53·91		68·6	231·8	146
..	60·00	..		44·98		35·47		43·67	34·5	28·1
265	220	203	208	203	210	234		180	144	287
Texas oil	Coal	Texas oil		Pocahontas coal		Welsh coal	Californian oil	Coal	Coal	Coal
19,000	13,800	19,000		15,000		14,800	18,500	14,200	14,000	14,800
Turbines	Turbines	..		Turbines		..	..	Recpro. turbines	Triple comp.	Triple comp.
8,600	5,000	..		612	3,484	..	..	1,550	2,773	1,696
8,810	9,250	1,145	3,800	1,365	5,549	2,021	1,590	1,920	5,934	3,138
..	74·0	..	..	11·48	46·73	37·41	29·49	26·24	25·60	21·49
1·110	1·23	0·287	0·950	0·255	1·037	1·057	0·831	0·641	0·742	0·765
1·025	1·85	..	..	2·23	1·60	..	..	1·24	2·14	1·85
105,500	67,500	15,750	45,410	14,547	46,686	15,450	14,983	18,786	50,607	28,460
11·92	7·30	13·75	11·94	10·66	8·87	7·51	10·73	9·79	8·53	9·09
14·00	8·58	16·70	14·50	12·47	10·37	9·16	13·09	11·76	10·34	10·63
13·24	9·00	3·93	11·35	2·76	8·72	8·08	7·84	6·27	7·67	6·94
12·27	13·50	..	..	23·77	13·42	...	..	12·12	18·25	16·82
0·92	1·50	..	..	8·74	1·54	...	..	1·93	1·35	2·41
71·7	60·6	84·9	73·7	80·3	66·8	59·5	68·3	80·0	71·3	70·8
13·84	12·83	..	..	24·27	24·27	..	..	..	..	41·35
12·76	19·26	..	..	212	37·4	..	..	..	..	103·4
1·05	1·43	..	..	8·79	2·78	..	..	..	..	6·16

Table XC.—Rates of Combustion and Evaporation. (Tank Boilers.)

Description of vessel.	Type of boiler, grate, &c.	Nature and force of draught.	Pounds of coal per ft. of grate per hour.	Pounds of water per pound of coal.	Pounds of water per ft. of grate per hour.
Ordinary Merchant Steamer .	Ordinary cylindrical, long bars	Chimney only .	15	9·15	137
,, ,, .	Ordinary cylindrical, short bars	,, ,,	20	9·25	185
Passenger or Mail Steamer .	Ordinary cylindrical medium bars . . .	Chimney, and ·5–·75 in. water	25–30	9–8·75	225–262
Cruiser or Battleship . .	Ordinary cylindrical, very long bars . . .	Chimney, and ·5 in. water .	30	8	240
,, ,, . .	,, ,,	Chimney, and 2 in. water .	42	7·75	325
Torpedo Gunboat . . .	Locomotive type, square grate	Chimney, and 2·5 in. water .	55	7·2	396
Torpedo-Boat . . .	,, ,, . .	3 in. water .	67	6·8	456
,, . . .	,, ,, . .	6 in. water .	96	6·6	633
Paddle Steamer . . .	Rectangular box, 7 foot bars	Chimney only	22	7·7	170
,, . . .	Rectangular box, 6 foot 6 inch bars, brass tubes .	Chimney, and ·5 in. water .	27	7·2	195
,, . . .	Ordinary cylindrical, 6 foot 6 inch bars, brass tubes .	Chimney, and ·5 in. water .	25	8·8	220

Table XCI.—Express Boilers. Particulars of Surface, Weight, &c.

Name of Ship.	Type of Boiler.	Inventor.	Number of Boilers.	Total Grate Area.	Total Heating Surface.	Ratio Heating Surface to Grate.	Working Pressure.	Total Weight of Boilers, etc.	Total Weight per sq. ft. of Grate.	Total Weight per I.H.P.	Total I.H.P. on Trial.	Air Pressure in Stokehold.	I.H.P. per sq. ft. of Grate.	I.H.P. per ton of Boilers.	Coal per I.H.P. per hour.	Coal per sq. ft. of Grate per hour.	Heating Surface per I.H.P.
				Sq.Ft.	Sq.Ft.		Lbs.	Tons.		Lbs.		Ins.			Lbs.	Lbs.	Sq.Ft.
H.M.S. :—																	
Rke.	Loco.	Laird	2	126	4,639	36·8	140	85·2	1515	70·0	2,740	2·30	21·7	32·1	..	..	1·69
Kta.	,,	Belliss	2	190	5,330	28·0	150	95·6	1127	55·7	3,840	1·70	20·2	40·4	..	..	1·3
Gmr.	,,	Sheerness	2	183	5,654	30·8	150	103·4	1267	63·4	3,654	2·70	19·9	35·4	..	..	1·5
Spy.	W. T.	Thornycroft	8	204	17,700	86·3	..	107·2	1177	51·0	4,703	1·70	23	43·9	..	..	3·7
Hvk.	Loco.	Yarrow	2	100	4,656	46·5	180	58·0	1299	37·0	3,497	3·66	34·9	60·0	2·59	90·6	1·33
Hnt.	W. T.	Yarrow	8	154·4	8,216	53·2	180	49·0	711	28·0	3,884	1·60	25·1	79·1	2·4	60·0	2·11
Dng.	,,	Thornycroft	3	189	8,892	41·7	210	56·0	644	30·0	4,208	3·37	22·2	75·0	3·49	77·8	2·11
Frt.	,,	Laird	4	154	8,812	52·7	175	64·2	900	32·0	4,484	4·10	29·9	69·7	3·88	113	1·96
		Normand															
Jns.	,,	Reed	4	209	10,164	48·6	210	70·6	757	41·0	3,885	2·39	18·5	54·9	3·15	58·6	2·61
Hgty.	,,	Yarrow	8	179	8,203	45·8	200	62·1	777	33·0	4,208	2·30	23·5	67·7	2·72	63·9	1·94
Cnlt.	,,	White	3	212·5	11,250	52·9	185	82·0	875	38·0	4,931	4·20	23·2	59·4	..	..	2·32
Strgn.	,,	Blechynden	4	176	10,022	57·0	200	62·2	791	32·0	4,367	3·1	24·8	70·1	3·29	81·6	2·29
Ql.	,,	Normand	4	210	11,945	56·8	220	75·0	800	26·0	6,347	4·25	30·2	84·6	2·48	74·9	1·88
Dspt.	,,	Thornycroft	3	196	11,855	60·5	220	66·0	764	25·0	5,795	3·45	29·5	87·8	2·37	70·1	2·04
Str.	,,	Reed	4	255	13,384	52·5	250	89·4	785	32·0	6,318	3·32	24·7	70·6	2·53	62·7	2·11
Gpy.	,,	Yarrow	4	245	13,561	55·3	250	68	622	23·0	6,523	4·4	26·6	95·9	2·51	66·8	2·07
An.	,,	Thornycroft	4	236	13,400	56·8	230	71	674	26·0	6,153	3·6	26·0	86·6	2·59	67·5	2·17
Al.	,,	White-Forster	6	750	45,000	60·0	220	208	801	20·0	30,000	..	40·0	112	1·86	74	1·50

Table XCII.—H.M.S. "Diana." Water Consumption Trials, showing Cost of Auxiliaries. Cylindrical S.E. Boilers. 155 lbs. Working Pressure. Made in May and June, 1899, for Admiralty Boiler Committee.

Description of Trial.	Percentage of Full Power.	Steam Pressure.			Lbs. of Water per I.H.P. per Hour.			Lbs. of Coal per I.H.P. per Hour.	Remarks.
		Boilers.	Engines, Starboard.	Engines, Port.	Main Engines.	Auxiliary.	All Purposes.		
30 hours at 800 I.H.P.	8·3	101	96	97	20·59	5·68	26·27	2·82	
12 ,, 1600 ,,	16·6	124	120	121	18·94	4·21	23·15	2·52	Cut-off in H.P. cylinder 38 per cent.
30 ,, 1600 ,,	16·6	117	113	113	18·26	3·29	21·55	2·52	Cut-off in H.P. cylinder 44·5 per cent.
30 ,, 4800 ,,	50·0	133	128	128	17·09	2·14	19·23	2·02	
30 ,, 6400 ,,	66·6	138	129	129	17·9	2·56	20·46	2·38	
8 ,, 8000 ,,	83·3	150	144	144	17·75	1·94	19·69	2·39	

Table XCIIa.—R.M.S. "Lusitania." Water Consumption Trials, showing the demands for Auxiliary Machinery and Domestic Purposes.

Speed in Knots.	15·75.	18·00.	21·00.	23·00.	25·40.
Shaft horse-power	13,400	20,500	33,000	48,000	68,850
Consumption per hour of steam of main engines lbs.	284,500	353,600	493,300	668,300	879,500
Consumption per hour of steam of auxiliaries, &c. lbs.	93,500	100,900	112,700	127,500	149,700
Consumption per hour of steam, total lbs.	338,000	454,500	606,000	795,800	1,029,200
Consumption per hour of steam of main engines per S.H.P. hour	21·23	17·24	14·91	13·92	12·77
Consumption per hour of steam of auxiliaries, &c., per S.H.P. hour	6·97	4·92	3·41	2·65	2·17
Consumption per hour of steam, total per S.H.P. hour	28·20	22·16	18·32	16·57	14·94
Feed water temperature, F.°	200	200	200	200	200
Consumption of coal per S.H.P. hour	2·76	2·17	1·80	1·62	1·46

Table XCIII.—Consumption of Water (Steam) and Fuel on Trials of Various Ships.

Ship.	Engines.		Boilers.		Per I.H.P. hour. Water for all purposes.			Per I.H.P. hour. Water Main Engines only.			Per I.H.P. hour. Fuel.		
	Sort of.	I.H.P.	Sort of.	Press.	Full Power.	⅘ to ⅔ Power.	Low Power.	Full Power.	⅘ to ⅔ Power.	Low Power.	Full Power.	⅘ to ⅔ Power.	Low Power.
Cargo s.s. . .	Triple	1,000	S.E. Cylindrical	..	17·50	18·52	18·81	..	..	..	..	..	..
,, . .	G. Turb.	1,000	,, ,,	..	14·75	16·06	17·28	..	..	..	..	..	..
Express s. . .	Turbines	65,000	D.E. Cylindrical	195	14·94	22·16	26·53	12·77	17·24	21·23	1·43	2·01	2·52
Passenger s. .	Mixed	7,000	,, ,,	195	13·66	13·79	..	11·95	12·80	13·60	..	..	..
Channel express .	Turbine	14,700	,, ,,	..	..	..	..	..	..	..	1·87	..	..
,, ,, .	Triples	7,000	D.E. & S.E. Cylr.	150	..	..	..	..	..	..	2·26	1·95	..
Cruiser B. . .	Triple	15,500	..	..	..	..	..	..	..	..	1·92	2·47	2·89
,, S. . .	Turbines	19,000	..	..	..	..	..	..	..	..	1·81	1·78	2·68
,, P. . .	,,	11,500	Water tube	246	14·9	15·50	21·8	..	..	..	1·82	..	..
,, Ch. .	Triple	23,500	Cylindrical	190	19·08	18·28	21·36	16·4	..	..	1·99	2·04	2·00
,, Jp. . .	Turbines	..	..	..	15·05	16·50	21·07	13·77	15·6	20·57	..	..	..
,, G. . .	,,	25,000	Cylr. & water tube	..	..	..	..	..	..	..	1·59	1·65	..
Battleship A. .	Triples	18,600	,, ,,	..	19·9	16·95	..	..	15·31	..	2·03	1·85	1·88
Cruiser Aa. . .	,,	16,800	Water tube	..	..	..	..	..	..	..	1·76	17·4	1·98
,, Hs. . .	,,	10,300	Belvl. water tube	228	..	..	..	..	..	..	1·467	1·54	..
,, A. . .	Turbines	14,700	Cylr. & water tube	..	..	..	..	..	..	..	..	..	..
Channel express .	,,	6,700	D.E. Cylindrical	160	17·1	..	..	..	..	..	1·71	..	..
,, ,, .	G. Turb.	5,000	,, ,,	160	14·3	..	..	..	..	..	..	..	..
Cargo s.s. . .	Triples	650	S.E. Cylindrical	180	13 35	..	..	13·35	..	..	1·46	..	..
Passenger 4-s.s. .	Mixed	10,800	,, ,,	180	15·17	12·37	..	..	10·65	..	..	..	..
Battleship . .	Turbines	40,500	Water tube	210	13·4	15·23	23·77	12·18	13·09	16·82	1·59	1·72	2·23

MARINE BOILERS—KINDS OF.

Boilers for ship purposes are divided into two classes known as tank and tube.

The tank boiler now used is cylindrical in form and contains within it the water and the apparatus for heating and evaporating it.

The tube boiler consists almost wholly of tubes, and generally there is a cylindrical vessel at the top, called the receiver, into which the tubes deliver the steam generated. If the tube delivers below the normal water level of the receiver they are said to be *drowned*, as distinct from those which deliver their contents above the water level, as was usual with the Thornycroft boilers. Tube boilers are also divided into two sub-divisions, each distinguished by the size of the tubes. Those made up to 1½-diameter are called the small tube variety, while those with tubes of greater diameter are called the large tube variety. The water is within the tubes, and the fire-place is an independent apparatus below the boiler; the heat is applied to the outer surface of the tubes.

The tank boiler contains much more water than the tube boiler, and is the one almost exclusively in use in all kinds of mercantile ships. In some few such ships, where very high speed and consequent very light weight is desirable, water tube boilers are used.

The water tube boiler is used very largely in all warships, as being lighter and safer against shot and shell. It is not, however, exclusively so, as at one time was the case. As a result of the investigations of the British Admiralty Boiler Committee, warships are now * supplied with about one-third of the power of cylindrical boilers, as they are better suited for and more economic when cruising at low speeds. The water tube boiler can be brought into action much quicker and stand rougher usage in the process than the tank, for steam can be raised in twenty minutes to half an hour after lighting fires on many water tube boilers, whereas several hours are necessary for raising steam in the tanks; in fact, it is not uncommon to allow six or more hours. There is very little difference in cost of a tube boiler for 300 lbs. working pressure over that of one for 150; a tank boiler of large size cannot be made for the former pressure, and for 230 lbs. is a much heavier and costlier one than if made for 150 lbs. On the other hand, the life of the tank boiler is *pari passu* longer than that of the water tube; and much longer than that of the express small tube boilers.

Cylindrical boilers are usually "horizontal," but there are one or two vertical forms used for other than auxiliary purposes with satisfaction. The Cochran boiler, and others similar to it, are useful and convenient for many purposes, especially where floor space is limited.

* Latterly there has been a reversion to the water tube boiler exclusively.

The Scotch haystack boiler was a very useful one for river steamers of high speed when moderate pressures were the rule.

The horizontal cylindrical boiler in general use has two, three, or four furnaces in the single-ended variety, and double those numbers in the double-ended ones of same diameters. They are now made up to 19 feet diameter, and the double-ended as much as 22 feet long. The working pressure is sometimes as high as 235 lbs., but generally 210 to 220 for quadruple expansion engines, and 180 to 205 for triples. In the Navy the pressure for cylindrical boilers is usually 190 lbs.

For such large diameters and high pressures, high tensile steel is generally used; 36 tons is the common test per square inch, while special steel having an ultimate strength of 40 tons can be used with advantage. There is no shock on such steel, no reversal of stresses, and the application and removal of load is most gentle.

The longitudinal seams are generally of the special treble-riveted type, but may be, as they are largely on the Continent and in America, special quadruple with vandyked edges to the butt straps, whereby as much as 94 per cent. of solid plate is obtained. The end circumferential seams are always double riveted for large sizes and high pressures, the middle seams of such double-ended ones are treble riveted.

The number of combustion chambers varies in practice, at one time even quite large boilers had sometimes only one chamber common to all six furnaces; sometimes a chamber common to all the furnaces of one end of a double-ended boiler. Sometimes, and this was quite general practice, there was a combustion chamber common to each pair of opposite furnaces, so that a six-furnace boiler would have three chambers. Of late years the tendency has been to have a separate chamber for each furnace: this is expensive and makes a heavy boiler, but is a safe one and one more easily worked well—especially at cleaning fire time than the others.

The gunboat boiler, in which the furnaces are at one end, the tubes at the other, and the combustion chamber between them, was largely used in the Navy; of late years it has been adopted with advantage in shallow passenger steamers where deck space is important and hold space free. It is a good steaming boiler and for small units compares very favourably with the ordinary cylindrical, but requires practically two stoke-holds, the second at the back to permit of cleaning or removing the tubes.

The locomotive boiler was formerly extensively used in small craft of all kinds, and in the Navy for very high powers. The water tube boiler has, however, quite superseded it.

The double-ended boiler is naturally the cheaper and lighter, as may be seen by referring to Table XCVIII. It is the one generally used for high powers in express steamers; but it requires more careful handling than similarly sized single-ended ones, especially when raising steam, as the bottom will remain quite cool after steam has formed on the top, unless means are adopted to ensure circulation.

Water tube boilers now in general use are much more limited in type and design than formerly. Time has tried them all, and com-

paratively few have continued to retain the confidence of marine engineers.

Of the large tube boilers there remains the Babcock & Wilcox, so much used in the British and American Navies; the Niclausse in the French Navy, and the Dürr in the German. The Belleville has practically gone, owing to having got a bad name—perhaps worse than it deserved,—but largely to its great cost and expense of upkeep. They all have straight tubes, are simple in design and construction, and easy to work. Their evaporative efficiency is good with good stoking, and they can be forced to a considerable amount without injury—especially the Niclausse, which has a positive circulation through each and every tube.

The Yarrow type of boiler is largely used in Naval ships of all sizes, and, whereas those in the big ships had tubes generally of $1\frac{1}{2}$ inches external diameter, they are now only $1\frac{1}{8}$ inches as in the smaller ships. Their tubes are nearly all straight, at an angle not very far from the vertical, and the general construction is simple, consequently they can stand being forced with rough usage, while being quite efficient evaporators.

The Hohenstein boiler also is one liked in the U.S. America; while in Japan the "Miyabara" boiler, somewhat like the Hohenstein in general design, is a favourite; both have straight tubes, and are simple in construction and easily worked.

Of small tube boilers there are, in addition to the Yarrow, the Normand and others of the same type, differing in detail. The double-ended variety of M. Sigaudy; the Mumford modification of it, the Blechynden variation from the Yarrow, and the White-Forster variation of the Blechynden are good boilers. Those of the Blechynden are good boilers, as are also the Mumford design, especially for small craft. The Admiralty continue to show a strong predilection for straight or nearly straight tubes for all kinds of water tube boilers; on the other hand, there is more scope for design, greater economy of space and flexibility of structure with the curved tubes, and with the moderate amount, as in the Normand type, little fault is to be found. The freer the flow in the tubes, the better both for efficiency and wear. With a strong positive flow there is no priming, and no danger from lodgment of bubbles nor any deposit of solids on the heating surface.

No doubt the troubles with the Belleville, and the controversies following them, largely account for the prejudice against all water tube boilers in the mercantile marine; there is, besides this, the fact that while these boilers are not necessarily wasteful of fuel, and are, when carefully worked, as economic as ordinary tank boilers, they can be made, in careless hands, more extravagant consumers than tank boilers can be under similar circumstances. On land, for central electricity stations, the water tube boiler is the favourite as responding to sudden and unexpected calls, and is found to be, with mechanical stoking, quite efficient and economical.

The total heating surface of a cylindrical marine boiler may be estimated with quite a fair degree of approximation by

Rule 245. Total heating surface $= D^2 \times L \times K$ square feet.

D is the diameter and L is the length of shell in feet. When the tubes are of moderate diameter ($2\frac{3}{4}$ to $3\frac{1}{2}$ inches) and spaced as usual in the mercantile marine, and with good room for examination and cleaning, K = 0·87 for single-ended and 1·08 for double-ended boilers. When the tubes are smaller ($2\frac{1}{2}$ to 3), and as many of them as can be got in consistent with satisfactory service, K may then be 1·0 for single and 1·2 for double-ended boilers.

Another method of obtaining roughly the total heating surface of a boiler is as follows:—

Rule 245a. Total heating surface $= D^2 \times L_1 \times \frac{6 \cdot 2}{d + 1 \cdot 5}$ sq. ft.

L_1 is the length of tubes in feet, and d their external diameter in inches. In case of a double-ended boiler, L_1 is twice the length of the tubes. A boiler may have 5 to 8 per cent. more surface than given by this rule when the steam space is restricted.

The weight of a cylindrical marine boiler can be ascertained with fair accuracy by the following:—

Rule 246. Weight of boiler $= \frac{D^2 \times L \times \sqrt{p}}{C}$ tons.

p is the pressure for which it is designed.

C is a factor, which, for single-ended boilers, is 768.

For double-ended boilers, with their combustion chambers common to opposite furnaces, C = 733.

For double-ended boilers, with separate chambers, C = 800.

Table XCIV.—Thermal Efficiency of various Boilers. Admiralty Boiler Committee, 1904.

Name of Ship.	Type of Boiler.	Duration of Trial.	Rate of Combustion.	Thermal Efficiency.	Remarks.
		Hours.	Lbs. per sq. ft. of Grate.	Per cent.	
Hermes,	Babcock & Wilcox,	30	20	81·0	Vertical baffles ; air over fires.
,,	,, ,,	29	27	77·8	,, ,, ,, ,,
,,	,, ,,	12	20·5	75·8	,, ,, no air ,,
,,	,, ,,	24	19	71·0	,, ,, ,, ,,
,,	,, ,,	30	27	70·3	,, ,, ,, ,,
Medea,	Yarrow (large tube),	26	18	75·7	
,,	,, ,,	8	40	69·5	
,,	,, ,,	24	17	75·0	
,,	,, ,,	24	21	75·0	
Hyacinth,	Belleville,	24·5	16	77·2	
,,	,,	11	20	73·3	
,,	,,	8	27	65·0	
Medusa,	Dürr,	8	35	64·8	
,,	,,	26	16	63·8	
,,	,,	24	18	61·7	
,,	,,	24	21	60·3	
Saxonia,	Cylindrical (Howden f. d.),	13	20	82·3	
Minerva,	,, (Admiralty f. d.),	25	14	69·7	
,,	,,	8·5	29	68·4	With retarders.
Sheldrake,	Babcock & Wilcox,	12	18	66	1898.
Espiegle,	,, ,,	9	13	73·2	1901.
Seagull,	Niclausse,	8	13	66·9	Maximum efficiency.
Fantôme,	,,	9	14	69·8	,, ,,

Table XCV.—Particulars of Destroyers built by Messrs Yarrow & Co. Ltd.

	H.M.S. "Hornet."	"Sokol."	"Mode."	H.M.S. "Welland."	H.M.S. "T.B.D."
Length between perpendiculars,	180 ft.	190 ft.	220 ft.	225 ft.	270 ft.
Breadth, extreme,	18 ft. 6 ins.	18 ft. 6 ins.	20 ft. 6 ins.	23 ft. 6 ins.	26 ft.
Displacement, tons	220	179	330	585	860
Engines, Kind of, { Twin screw	{ 3 cylinder triples	3 cylinder triples	4 cylinder triples. Balanced Y.S.T. system	4 cylinder triples. Balanced Y.S.T. system	Turbines
Cylinders, Diameter and Stroke,	18–26–39½ / 18	18–26–39½ / 18	20½–31½–34–34 / 18	23–35–38–38 / 20	Compd. turbines
Boilers, Kind of, and No.,	Yarrow straight tube 8	Yarrow straight tube 8	Yarrow straight tube 4	Yarrow straight tube 4	Yarrow straight tube 2
Steam pressure,	170	160	241	232	Oil fuel
Grate area, square feet,	172	172	220	270	...
Total heating surface, sq. feet,	8320	8320	13,000	16,000	16,600
Air pressure,	1½"	1 5/16"	1·4"	1·7"	...
Coal per sq. ft. grate, per hour, lbs.	55·34	45·0	68·0	47·0	...
Coal per I.H.P., lbs.	2·45	2·1	...	1·63	Oil 1·033
Revolutions per minute,	392·65	405	421·4	342·3	...
Mean total I.H.P. at full speed,	3884	3800	...	7766	17,900
Speed, mean, knots	{ 27·628 for 3 hours	29·762 for 3 hours	32·428 for 3 hours	26·244 for 4 hours	35

Table XCVI.—Comparison of Water Tube Boilers with Others in various Ships (Trial Trips).

Class of Ship.	Battleships.		First-Class Cruisers.		Second-Class Cruisers.			Third-Class Cruisers.		Torpedo Gunboats.			
Name.	Mean of Six.	Mean of Six.	"Edgar" Class.	"Diadem" Class.	"Diana" Class.	"Apollo" Class.	"Arrogant" Class.	"Pearl" Class.	"Pelorus" Class.	"Alarm" Class.	"Speedy."	"Shel-drake."	"Seagull."
Indicated horse-power,	12,414	13,500	12,851	16,961	9,846	9,274	10,240	7,469	7,152	3,665	4,703	4,050	3,546
Boiler pressure,	149	289	146·2	286·5	151	155	265	155	253	155	..	151	145
Air pressure in stokehole,	0·85	0·19	0·7	0·305	1·06	0·9	0·06	1·4	2·9	2·20	1·70	0·5	0·16
Type of boiler,	Cylin-drical.	Belle-ville.	Cylin-drical.	Belle-ville.	Cylind. S.E.	Cylind. D.E.	Belle-ville, old.	Cylind. D.E.	Ex-press.	Loco-motive	Thorny-croft.	Bab-cock.	Ni-clausse.
Grate area, sq. ft.	817	1,050	812	1,460	624	575	869	410	353	171	204	252	276
Total heating surface, ,,	25,233	33,700	24,908	40,990	18,579	15,641	25,606	11,025	20,508	6,204	17,700	9,103	7,932
Total weight of boilers, &c., tons	724	623	646	767	548	434	463	314	172	124·4	132	124·8	134·7
Ratio heating surface to grate,	30·8	32·1	30·7	28·1	29·8	27·2	29·4	26·9	58·0	36·3	86·8	37·4	28·8
Coal per I.H.P. per hr., lbs.	2·29	1·72	1·70(*a*)	1·85	2·18(*b*)	..	2·00 (*c*)	..	2·36(*d*)	..	..	1·57	2·11
,, sq. ft. grate per hour, ,,	34·8	..	..	..	..	..	..	..	..	..	..	25·6	27·1
Weight of boilers, &c., per I.H.P., ,,	131	103	111	101	124	104	99	98·6	52·0	76·0	62·9	69·9	85·1
Weight of boilers, &c., per sq. ft. of grate, ,,	1,971	1,329	1,687	1,172	1,959	1,674	1,170	1,715	1,092	1,630	1,450	1,109	1,093
I.H.P. per sq. ft. of grate,	15·2	12·9	15·2	11·6	15·79	16·1	11·82	18·2	20·9	21·4	23·0	16·0	12·8
,, ton of boiler,	17·1	21·7	20·1	22·1	17·97	20·7	22·5	23·8	43·1	29·5	39·6	32·5	26·3
Total heating surface per I.H.P.,	2·0	2·5	1·9	2·41	1·88	1·65	2·49	1·47	2·87	1·70	3·7	2·3	2·24
Total weight of machinery, tons	1,341	1,290	1,161	1,540	915	740	825	539	383	211	212	205	214
I.H.P. per ton of machinery,	9·3	10·5	11·1	11·0	10·76	12·5	12·34	13·8	18·92	17·3	22·1	19·3	16·9
Piston speed, ft.	875	918	923	923	937	910	906	880	981	875	875	875	..
Coal consumption per I.H.P. at sea trial,	2·0	2·36	2·0	2·33	2·11	..	2·36	..	..	..	..	2·14	2·15
I.H.P. on sea trial, 1 ship,	8,029	8,163	8,824	13,517	6,694	..	6,273	..	..	..	..	2,775	2,773

(*a*) At 10,517 I.H.P., (*b*) at 8307 I.H.P., (*c*) at 7275 H.P., (*d*) at 5388 H.P.

EVAPORATION, HEATING SURFACE, &c.

The efficiency of the heating surface of a boiler depends on its position, the material and thickness of the plates, tubes, etc., the condition of the surfaces, and the circulation of the gases on the one side, and water in contact with them on the other, and their difference in temperature.

(1) **Position.**—The most efficient heating surface is that of the furnace crowns and combustion chambers; its high efficiency is due partly to the great difference of temperature between the two sides of the plates, partly to the freedom of the surfaces from deposit of soot or ash, and also in some measure to the fact that mineral or earthy matter deposited from the water does not readily adhere to vertical surfaces or to surfaces over which there is a good circulation of water.

Next in order of efficiency come the upper surfaces of the tubes, more especially of the *upper* tubes at the ends next the combustion chambers, for the reason that the flame and heated gases always seek the highest possible course and rapidly lose heat in passing through the tubes, entering them at, say, 2400° and leaving them under 800°, and also because the bubbles of steam disengage themselves more easily when they can rise directly from a surface.

When a "nest" of tubes is made very deep (*i.e.* when there are many rows one above another) it very often happens, for the reason just given, that the lower rows do little or no work, and might be often better absent from the boiler.

The surfaces below the level of the fire-bars, and the front tube-plates are generally left out of consideration as of no practical value.

(2) **Material.**—Amongst ordinary metals suitable for boiler construction, copper is by far the most efficient conductor of heat, its conductivity compared with iron or steel being about as 3 to 1; for various reasons, however, it is now very rarely used, and even brass boiler tubes are now hardly ever used. Steel for the shells, furnaces, and combustion chambers and for the tubes of water tube boilers, and iron and steel for the tubes of cylindrical boilers, are now the universal materials.

The high conductivity of copper so reduces the possible thermal difference between the fire side and the water side of a plate as to very materially diminish the local racking strains ordinarily produced during the operation of getting up steam, and for this reason it was peculiarly suitable for the fire boxes of torpedo-boat or other boilers in which the changes of temperature are sudden and great.

(3) **Condition of surfaces.**—A perfectly smooth and clean metallic surface does *not* give satisfactory results as a heating surface, as it is apt to cause sudden and violent intermittent ebullition, and, for this reason, few boilers work quite satisfactorily until they have become slightly corroded or have acquired a thin film of scale or deposit.

Since the cooling of the products of combustion takes place mainly in the tubes, soot is principally deposited in them, and they must be swept more or less frequently, according to the class of coal used, if the efficiency of the surface is to be maintained.

The external surfaces of the tubes, especially on the upper sides, are peculiarly liable to the accumulation of deposit where water containing salt or other mineral matter is used, as there is rapid evaporation, and the upward circulation is naturally sluggish and much impeded in large "nests" of tubes.

(4) **Circulation.**—The proper circulation of the water over the heating surfaces is of the very greatest importance, for no boiler in which the design and arrangements are such as to prevent such a circulation will ever steam satisfactorily or evaporate the quantity of water that it could do otherwise. The principal evils that result from defective circulation are deposit of scale on the evaporating surfaces, with the consequent overheating and buckling of plates and leaking of tubes, and the irregular and much diminished evaporation, accompanied by intermittent fits of "priming."

Special difficulty was found in getting a proper circulation of water over the tube plates next the fires of Naval boilers, mainly on account of the close pitch of the tubes and consequent contraction of the waterways; and similar difficulty was also experienced with the large flat (almost square) tops of the fire-boxes in boilers of the locomotive type.

In this latter case, the cure was found by fitting one or two "Galloway" tubes, a cure as effectual as was found many years ago to be the case in large double-ended cylindrical boilers. Better results, however, were obtained by dividing the fire-box completely by means of a water space, thus making two distinct furnaces.

Diameter of tubes.—The diameter of tube has a considerable influence on the efficiency of its surface, inasmuch as the contents increase as the square, whilst the surface increases as the diameter only. Thus if a 4-inch tube be substituted for two 2-inch ones, the absorbing surface will remain the same, but twice the quantity of gas will be passed, if the velocity of flow remain the same. If the velocity of flow be reduced in the 4-inch tube (to one half) so as to pass only the same quantity of gas as the two 2-inch ones, the 4-inch tube is still at a disadvantage,—inasmuch as the mean distance of the molecules of gas from its surface is greater than in the 2-inch ones. Howden's retarders somewhat remedied this defect.

When artificial or forced draught is used, the diameter of tube should be smaller than for natural draught,—if length of tube and velocity of gas are to remain the same,—since the gases have a higher temperature and must be divided into smaller threads or streams to prevent escape into the chimney at a wastefully high temperature: the draught, being

positive and sharp, may be relied on to overcome the extra friction as well as to prevent the deposit of soot.

The diameters of tube in ordinary use are as follows:—

(*a*) Merchant steamer.	Chimney draught.	3 inch to 3½ inch.
(*b*) ,, ,,	Artificial ,,	2½ inch to 2¾ inch.
(*c*) Cruiser or battleship.	,, ,,	2½ inch, cyl. boilers.

Quantity of water evaporated per pound of coal.—This of course depends not only on the quality of the coal, but on the type of boiler, the strength of draught, and the skill of the stoker. The average evaporation of water that may be expected with good fuel, under the various conditions indicated, is shown in Tables LXXXIX., XCIII., &c. A good cylindrical boiler should have an efficiency of 80 per cent. at least.

Equivalent evaporation from and at 212° F.—In order to compare evaporative results obtained with different temperatures of feed and pressures of steam, it is necessary to eliminate the effects of the varying conditions by reducing all the results to one common standard.

The standard generally employed is "the equivalent evaporation from and at 212° F.,"—*i.e.* the number of pounds of water that would be evaporated in each case, per pound of coal per hour, if the feed water were supplied at 212° F., and completely evaporated under the pressure (one atmosphere) due to that temperature.

On reference to the Table "Properties of Saturated Steam," it is seen that 966 thermal units are required to evaporate 1 lb. of water from 212°, and under a pressure of one atmosphere; so that, if the number of thermal units imparted to the water by each pound of coal during the test be determined, and divided by 966, the quotient will be the equivalent evaporation from and at 212°. The quantity of heat imparted by each pound of coal is determined by referring to column 7 of the same Table, opposite to the proper pressure, and there reading off the total heat, from 32°, contained in each pound of steam, deducting from this the number of units, above 32°, contained in each pound of feed-water as supplied, and then multiplying the remainder by the number of pounds of water evaporated, per pound of coal per hour, during the test. This operation is concisely expressed by the formula,—

$$N_{212} = N_T \left\{ \frac{H - (t - 32)}{966} \right\} \text{ (\textit{vide} Table C., p. 337)} \cdot$$

where N_{212} = Number of lbs. of water evaporated per lb. coal, from and at 212°.

N_T = Number of lbs. water evaporated per lb. coal under test conditions.

H = total heat, from 32°, per lb. of steam at test temperature and pressure

t = temperature at which feed was supplied during test.

Table XCVII.—Leading Particulars of some Boilers made since 1895.

Design-nation.	Made in Accordance with the Rules of	Working Pressure.	Shell.			Furnaces.			Number of Combustion Chambers.	Tubes.				Total Heating Surface.	Weight of			Weight per 100 feet of Total Heating Surface.			Sq. Ft. Total H.S. per Ton of Boiler.
			Diameter.	Length.	Thickness.	Number.	Diameter.	Thickness.		Number.	Diameter.	Length.	Surface.		Boiler.	Water.	Total.	Boiler.	Water.	Total.	
		Lbs.	Ft. In.	Ft. In.	In.		In.	In.			In.	Ft. In.	Sq. Ft.	Sq. Ft.	Tons.	Tons.	Tons.	Tons.	Tons.	Tons.	
H D	Board of Trade,	190	16 5	17 10	$1\frac{11}{32}$	8	$40\frac{3}{8}$	$\frac{13}{32}$	...	848	$2\frac{3}{4}$	7 2	4293	5112	83·0	47	130	1·62	0·92	2·54	61·6
W F	Board of Trade and Lloyd's,	180	16 4	17 6	$1\frac{1}{2}$	8	$40\frac{5}{8}$	$\frac{9}{16}$	...	840	$2\frac{1}{2}$	7 $0\frac{1}{2}$	4170	4572	79·5	45·5	125	1·74	0·99	2·73	57·5
P R	Board of Trade,	210	15 6	19 6	$1\frac{13}{32}$	6	44	$\frac{11}{16}$	...	852	$2\frac{5}{8}$	7 $10\frac{5}{8}$	4540	5210	83·0	46·75	129·7	1·59	0·91	2·50	62·7
W N	,, ,,	192	15 3	18 6	$1\frac{13}{32}$	6	$43\frac{1}{2}$	$\frac{5}{8}$	...	826	$2\frac{5}{8}$	7 6	4180	4839	73·75	42·75	116·5	1·52	0·88	2·40	65·6
B T	,, ,,	214	14 5	19 10	$1\frac{5}{16}$	6	$40\frac{1}{16}$	$\frac{11}{16}$	...	740	$2\frac{1}{2}$	7 11	3760	4415	75·5	41·75	117·25	1·71	0·95	2·66	58·5
C A	Board of Trade and Lloyd's,	200	14 3	19 6	$1\frac{13}{32}$	6	42	$\frac{5}{8}$	3	730	$2\frac{1}{2}$	7 $7\frac{1}{2}$	3578	4575	68·5	40·5	109·0	1·50	0·88	2·38	66·8
I D	Board of Trade,	216	14 2	19 6	$1\frac{9}{16}$	6	$40\frac{1}{16}$	$\frac{11}{32}$	...	612	$2\frac{3}{4}$	8 0	3460	4071	72·5	40·5	113·0	1·78	1·00	2·78	56·1
C P	Lloyd's,	180	12 9	18 0	$1\frac{1}{16}$	6	38	$\frac{11}{32}$	3	616	$2\frac{3}{4}$	6 6	2826	3415	41·0	30·1	71·1	1·20	0·88	2·08	83·3
P E	Board of Trade and Lloyd's,	190	11 6	18 9	$1\frac{5}{32}$	4	42	$\frac{13}{32}$	4	496	$2\frac{3}{4}$	7 $1\frac{5}{8}$	2494	3000	43·2	22·5	65·7	1·44	0·75	2·19	69·4
P G	Board of Trade and Lloyd's,	180	15 0	11 9	$1\frac{11}{32}$	3	45	$\frac{13}{32}$	3	409	$2\frac{5}{8}$	8 0	2355	2860	45·0	26·0	71·0	1·57	0·91	2·48	63·5
O O	Board of Trade and Lloyd's,	200	14 3	11 6	$1\frac{5}{16}$	3	44	$\frac{5}{8}$	3	238	$3\frac{1}{2}$	7 9	1667	2138	40·6	19·0	59·6	1·90	0·89	2·79	52·6
D N	Board of Trade,	160	14 3	11 6	$1\frac{3}{16}$	3	44	$\frac{11}{32}$	3	240	$3\frac{1}{2}$	8 $1\frac{1}{2}$	1756	2232	37·85	19·55	57·4	1·70	0·88	2·58	56·3
Z O	Board of Trade and Lloyd's,	200	13 0	11 0	$1\frac{1}{16}$	3	38	$\frac{9}{16}$	3	234	$3\frac{1}{4}$	7 6	1492	1907	34·8	17·2	52·0	1·82	0·90	2·72	54·8
S A	Lloyd's,	180	13 0	10 6	$1\frac{1}{16}$	3	46	$\frac{13}{32}$	3	176	$3\frac{1}{2}$	6 9	1067	1335	26·5	16·5	43·0	1·92	1·24	3·16	50·4
D O	Board of Trade and British Corporation,	200	11 6	11 8	$1\frac{1}{8}$	2	40	$\frac{13}{32}$	2	172	$3\frac{1}{4}$	7 7	1088	1327	27·3	13·8	41·1	1·91	1·04	2·95	50·0 serve tubes
L Z	Lloyd's,	200	11 0	9 6	1	2	$37\frac{3}{4}$	$\frac{9}{16}$	2	132	$3\frac{1}{4}$	6 3	686	880	18·85	10·5	29·35	2·14	1·19	3·33	46·7
B Y	,,	160	11 0	9 6	$\frac{13}{16}$	2	36	$\frac{7}{16}$	2	146	$3\frac{1}{4}$	6 3	761	1000	17·6	10·2	27·8	1·76	1·02	2·78	56·8
D N	,,	200	10 0	9 6	1	2	35	$\frac{13}{32}$	2	118	$3\frac{1}{2}$	6 $5\frac{1}{2}$	650	820	17·3	8·75	26·05	2·11	1·67	3·78	47·4
C O	,,	180	10 0	9 6	$\frac{13}{16}$	2	35	$\frac{9}{16}$	2	118	$3\frac{1}{2}$	6 $5\frac{1}{2}$	650	820	16·4	8·75	25·15	2·00	1·67	3·67	50·0
L N	Bureau Veritas,	180	8 9	9 9	$\frac{27}{32}$	2	32	$\frac{11}{16}$	1	82	$3\frac{1}{2}$	6 9	561	600	13·4	7·45	20·85	2·23	1·24	3·47	44·8

PROPORTIONS OF BOILERS.

Grate surface.—From the information given in Tables LXXXIX. and XCI., it will be an easy matter to determine the I.H.P. per foot of grate that may be anticipated (or the grate area that must be provided for any given I.H.P.) in the case of proposed new boilers of any of the types mentioned.

Size of furnace.—The furnaces of cylindrical boilers should not be less than 30 inches in diameter, nor more than 48 inches, unless in exceptional cases: wherever possible a diameter of not less than 40 inches should be given, as, with smaller furnaces,—owing to thickness of fire being practically constant for all diameters,—the space above the fuel is much contracted, and the combustion less perfect in consequence. As previously stated, long bars cannot be properly worked by hand, and are not so efficient as short ones: it is desirable that they should be limited to a length of 5 feet.

Number of furnaces.—

Boilers up to	9 feet diameter	may have	1 furnace.
,, ,,	13 ft. 6 ins.	,,	2 furnaces.
,, ,,	15 feet	,,	3 ,,
,, beyond	15 feet	,,	4 ,,

If the boilers are double-ended, the number of furnaces will, of course, be double the figure given above.

The total heating surface necessary and sufficient for each ship can be only determined by the special circumstances. It will be seen by reference to Table LXXXIX. that 10 to 13 lbs. of steam can be got per square foot of heating surface in water tube boilers, and as much as 9 lbs. from cylindrical; in both cases the draught is forced and the surfaces within and without quite clean but the efficiency is low for economic working. In a general way it may be taken that 8 lbs. per square foot of T.H. surface may be obtained economically with clean boilers; and to provide for dirty tubes, &c., 7 lbs. is as much as should be counted on with ships making a long enough voyage to cause fouling. For short voyages not exceeding eight hours and where economy of fuel is not of first importance, 8 may be taken. Assuming that the steam consumption on every kind of ship, including that of the engine's own auxiliaries (that is, air and circulating and feed pumps, etc.), is as follows:—

Turbines, best modern geared	.	10·5 lbs.	per H.P. hour.
Quadruples generally . .	.	14·0 ,,	,,
Triples, best economic . .	.	14·7 ,,	,,
,, express service . .	.	15·5 ,,	,,
Naval ships turbines (direct)	.	13·5 ,,	,,
,, reciprocators .	.	16·0 ,,	,,

Table XCVIII.—Particulars of Modern Cylindrical Boilers (Scantlings).

	A.	B.	C.	D.	E.	F.	G.	H.
Diameter, mean,	17′ 8″	15′ 3″	16′ 4″	16′ 2″	16′ 3″	15′ 6″	16′ 0″	15′ 3″
Length over plates,	22·0	18′ 6″	17′ 6″	11′ 0″	10′ 9″	11′ 0″	11′ 6″	12′ 0″
Working pressure, lbs.	195	230	180	220	190	215	180	180
Rules on which designed,	B.T.	B.T.	B.T. B.V.	B.T. Ll.	Admlty.	B.T. Ll.	B.T.	B.T.
Number of furnaces,	8	6	8	4	4	3	3	3
Diameter of furnaces, ins.	49·5	$46\frac{1}{4}$	$42\frac{1}{2}$	$45\frac{1}{2}$	$46\frac{3}{4}$	$49\frac{1}{4}$	52	49
Thickness of furnaces, ins.	0·655	$\frac{21}{32}$	$\frac{19}{32}$	$\frac{23}{32}$	$\frac{19}{32}$	$\frac{23}{32}$	$\frac{21}{32}$	$\frac{20}{32}$
Number of combustion chambers,	8	3	4	4	2	3	3	3
Number of tubes,	1064	644	840	438	578	396	342	463
Diameter of tubes, ins.	$2\frac{1}{2}$	$3\frac{1}{4}$	$2\frac{3}{4}$	$2\frac{3}{4}$	$2\frac{1}{2}$	$2\frac{3}{4}$	$3\frac{1}{4}$	$2\frac{1}{2}$
Length of tubes,	8′ 0½″	7′ 4″	6′ 11″	7′ 8″	7′ 5½″	7′ 2¼″	7′ 9″	7′ 6″
Surface of tubes, sq. ft.	5480	4017	4170	2417	2820	2044	2254	2272
Surface of other parts, sq. ft.	1120	669	861	544	467	462	516	465
Total heating surface, sq. ft.	6600	4686	5031	2961	3287	2506	2770	2737
Grate area, sq. ft.	169	121	160	90	94·2	...	72	67·8
Thickness of shell, ins.	1·60	$1\frac{7}{16}$	$1\frac{1}{2}$	$1\frac{21}{32}$	$1\frac{17}{32}$	$1\frac{21}{32}$	$1\frac{7}{16}$	$1\frac{11}{32}$
Diameter of shell rivets, ins.	1·63	$1\frac{9}{16}$	$1\frac{1}{2}$	$1\frac{21}{32}$	$1\frac{17}{32}$	$1\frac{21}{32}$	...	$1\frac{7}{16}$
Pitch of shell rivets, ins.	10·5	9″	10″	10″	10″	10″	...	$9\frac{11}{16}$″
Tensile test shell plates,	...	36–40	26	31–34	28–31	29–32	29–32	29–32
Thickness of top end plates,	...	$1\frac{1}{2}$	$1\frac{1}{16}$	$1\frac{1}{4}$	$1\frac{1}{16}$	$1\frac{1}{4}$	$\frac{7}{8}$	$\frac{7}{8}$
,, front tube plates,	0·875	1	$\frac{7}{8}$	$\frac{7}{8}$	$\frac{7}{8}$	$\frac{7}{8}$	$\frac{7}{8}$	$\frac{7}{8}$
,, back tube plates,	0·812	...	...	$\frac{7}{8}$	$\frac{7}{8}$	$\frac{7}{8}$	$\frac{7}{8}$	$\frac{7}{8}$
,, combustion chambers,	0·655	$\frac{23}{32}$	$\frac{19}{32}$	$\frac{5}{8}$	...	...	$\frac{11}{16}$	$\frac{5}{8}$
Weight as finished, tons	...	...	78·5	57·0	52·0	53·0	...	...
Weight per 100 sq. ft. T.H.S., tons	...	...	1·560	1·926	1·582	2·115	...	...

To each case add the amount of steam required for other auxiliaries and domestic purposes, and the proper allowance of T.H. surface is obtained by dividing the sum by 7 or 8 as the case may be.

Of course each case may have a special rate of steam consumption of the main engines, and that should be taken instead of the average figures as above. The allowance for auxiliaries and domestic is also a variable quantity, but may be taken as 5 per cent. in an ordinary cargo steamer and may be as much as 15 per cent. on a passenger steamer in the North Atlantic or North Sea for winter service. With these assumptions the following holds good:—

Table XCIX.—Total Heating Surface per I.H.P. of various Ships.

Description of Ship and Service.	Steam Consumption.			Total Heating Surface per H.P.
	Main.	Auxly.	Total.	
				Sq. ft.
(1) Cargo steamer, general service, quadruple reciprocators . .	14·0	0·70	14·70	2·100
(2) Cargo steamer, general service, triple reciprocators . . .	14·7	0·74	15·44	2·206
(3) Passenger steamer, express long voyage, triple reciprocators .	15·5	2·00	17·50	2·500
(4) Passenger steamer, express short voyage, triple reciprocators .	15·5	1·50	17·00	2·125
(5) Passenger and cargo steamer, long voyage, quadruple reciprocators	14·0	1·20	15·25	2·180
(6) Atlantic passenger, express, turbine-driven	12·5	2·50	15·00	2·143
(7) Atlantic passenger and cargo, turbine-driven	12·5	1·50	14·00	2·000
(8) Short service, express, cold climate, turbine-driven	13·0	1·65	14·65	1·88
(9) Passenger and cargo steamer, geared, turbine-driven . .	10·5	2·3	12·8	1·83
(10) Naval ships, battleships, and large cruiser turbines	13·5	2·00	15·50	1·925
(11) Naval ships, battleships, and large cruiser reciprocators . . .	16·0	2·00	18·00	2·25
(12) Naval ships, scouts, and high-speed turbines	14·0	2·00	16·00	2·00

Taking in each case the maximum average power developed by the

engine during a run of not less than two hours as the basis, the allowance on the sea full power will be larger than above.

Steam room.—The steam room allowed in the various types of boiler, working at or about the pressures named, may be as follows:—

Table XCIXa.—Allowance of Steam Room in Boilers (Cubic Feet per I.H.P.)

Description of Boiler.	Type of Engine.	Working Pressure.	Allowance.
		Lbs.	Cubic feet
(1) Cylindrical S. and D. ended, ordinary .	Triple and quadruple screw .	200–230	0·36–0·32
(2) Cylindrical S. and D. ended, ordinary .	Triple screw, slow revolutions .	175–200	0·38–0·35
(3) Cylindrical S. and D. ended, ordinary .	Triple screw, high revolutions .	175–200	0·33–0·28
(4) Cylindrical S. and D. ended, ordinary .	Compound paddle wheel . .	120–140	0·43–0·38
(5) Cylindrical S. and D. ended, ordinary .	Turbines . .	160–180	0·24–0·18
(6) Water tube as used in H.M. Navy . .	,, . .	180–200	0·085–·065
(7) Water tube as used in H.M. Navy . .	Triple screw, very high revolutions	210–250	0·095–·08

In cylindrical boilers the top row of tubes should be not less than ·28 × diameter of boiler from the top; if higher, the contraction of water surface is apt to cause priming.

Paddle and all slow-running engines should have considerably larger steam spaces than the faster running ones, but in practice weight of machinery is of such great importance that everything is cut down to the lowest limit; so, many compound paddle engines, working at pressures of 100 to 120 lbs., have not more than ·4 cubic foot per I.H.P.

Water spaces, &c.—The spaces between the furnaces themselves, between the furnaces and the shell, and between the combustion chambers, although sometimes made as narrow as 5 inches, are better 5½ inches, and even 6 inches when possible; the space between back of combustion chamber and end of boiler should be 6 inches at the bottom, increasing to 9, 10, or even 12 inches at the top.

A suitable pitch for tubes is 1·35 to 1·4 × external diameter of tube; where weight is of great importance, as in Naval vessels, the tubes are sometimes pitched a little closer than this, but with water spaces so

contracted there is always risk of priming. A fair average pitch is given by the rule,—

Rule 247. Pitch of tubes $= D + \frac{\sqrt{D}}{2} + .35$ inch;

where D is outside diameter of tube in inches.

Whenever possible the tubes should be placed in horizontal and vertical rows, and not arranged in any diagonal lines or zigzag fashion.

The clear space between the nests of tubes should never be less than 10 inches, and when possible should be 10½ or 11 inches,—in order that a man may be able to get down: manholes between the furnaces and tubes can be then dispensed with—an important point,—for the life of a boiler depends to a considerable extent on the number of openings in its shell,—the leakage from such openings being most destructive. For the same reason, longitudinal seams below the water level should be avoided as much as possible

Tables XCVII. and XCVIII. give particulars (dimensions and weights) of a number of boilers that have been actually constructed in accordance with different rules as indicated. The weights given are those of "bare boiler," without either furnace fittings or boiler mountings.

Table C.—Multipliers for Converting Weight of Water actually evaporated to the Equivalent Quantity from and at 212° F.

Temp. of Feed Water.	Boiler pressure, being absolute pressure minus 14·7 lbs.														
	110	120	130	140	150	160	170	180	190	200	210	220	230	240	250
F°															
60	1·200	1·202	1·203	1·205	1·207	1·208	1·210	1·211	1·212	1·214	1·215	1·216	1·217	1·218	1·219
70	1·189	1·191	1·193	1·194	1·196	1·197	1·199	1·200	1·202	1·203	1·205	1·206	1·207	1·208	1·209
80	1·179	1·181	1·183	1·184	1·186	1·187	1·189	1·190	1·192	1·193	1·194	1·195	1·196	1·198	1·199
90	1·169	1·170	1·172	1·174	1·176	1·177	1·179	1·180	1·181	1·183	1·184	1·185	1·186	1·187	1·188
100	1·158	1·160	1·162	1·164	1·165	1·167	1·168	1·170	1·171	1·172	1·174	1·175	1·176	1·177	1·178
110	1·148	1·150	1·152	1·153	1·155	1·156	1·158	1·159	1·160	1·162	1·163	1·164	1·166	1·167	1·168
120	1·138	1·140	1·141	1·143	1·145	1·146	1·147	1·149	1·150	1·151	1·153	1·154	1·155	1·156	1·157
130	1·127	1·129	1·130	1·132	1·134	1·136	1·137	1·138	1·140	1·141	1·142	1·144	1·145	1·146	1·147
140	1·117	1·119	1·120	1·122	1·124	1·125	1·127	1·128	1·129	1·131	1·132	1·133	1·134	1·135	1·136
150	1·106	1·108	1·110	1·111	1·113	1·115	1·116	1·118	1·119	1·120	1·121	1·123	1·124	1·125	1·126
160	1·096	1·098	1·100	1·101	1·103	1·104	1·106	1·107	1·108	1·110	1·111	1·112	1·113	1·115	1·116
170	1·085	1·087	1·089	1·091	1·092	1·094	1·095	1·097	1·098	1·099	1·101	1·102	1·103	1·104	1·105
180	1·075	1·077	1·079	1·080	1·082	1·083	1·085	1·086	1·088	1·089	1·090	1·091	1·092	1·094	1·095
190	1·065	1·066	1·068	1·070	1·071	1·073	1·074	1·076	1·077	1·078	1·080	1·081	1·082	1·083	1·084
200	1·054	1·056	1·058	1·059	1·061	1·063	1·064	1·065	1·067	1·068	1·069	1·071	1·072	1·073	1·074
210	1·044	1·046	1·047	1·049	1·051	1·052	1·053	1·055	1·056	1·057	1·059	1·060	1·061	1·062	1·063

Table CI.—Relative Weights of various Boiler Installations. Admiralty Boiler Committee, 1904.

Name of Ship.	Type of Boiler.	Weight of Boiler Installation.	Maximum Output Steam per Hour.	Output per Hour per Ton of Boiler.	Remarks.
		Tons.	Lbs.	Lbs.	
Saxonia, .	Cylindrical (Howden), .	1000	132,600	132·6	
Minerva, .	,, (Admiralty f. d.),	567	167,100	295	With Retarders.
,,	,, ,,	556	156,200	280	Without ,,
Hyacinth,	Belleville, ...	454	178,700	394	
Medea, . .	Yarrow, ...	330	157,800	478	
Medusa, .	Dürr, ...	314	158,000	503	
Hermes, .	Babcock & Wilcox, . .	490	200,000	410	With baffles and air over fires.
,,	,, ,, . .	481	182,300	380	As originally fitted, 1903.
Sheldrake,	,, ,, . .	125	43,840	351	1898.
Espiegle, .	,, ,, . .	95	24,780	261	1901.
Seagull, .	Niclausse, ...	135	28,450	359	
Fantôme, .	,, ...	76·5	24,750	297	

Table CIa.—Comparison of Steel Boilers for 160 lbs. pressure by Board of Trade Rules (prior to 1895).

General Description.	No. of Boilers.	Shell.		Furnaces.			Tubes.				Total Heating Surface.	Total Capacity.	Weight.		
		Diam.	Lgth.	No.	Diam.	Lgth.	No.	Diam.	Lgth.	Surface.			Boiler	Water	Total.
		Ft. in.	Ft. in.		Ins.	Ft. in.		Ins.	Ft. in.	Sq. ft.	Sq. ft.	C. ft.	Tons.	Tons.	Tons.
Double-ended boiler, having two combustion chambers, each common to *opposite* furnaces, . .	1	12 0	14 0	4	41	5 6	376	3½	5 6	1715	2000	1582	35·3	21	56·3
Double-ended boiler, having two combustion chambers, each common to furnaces at *same* end, . .	1	12 0	14 0	4	41	5 3	376	3½	5 3	1630	2000	1582	36	21	57
Double-ended boiler, having four combustion chambers, one to each furnace, ...	1	12 0	14 0	4	41	5 2	376	3½	5 2	1610	2000	1582	36·4	21·5	57·9
Single-ended boiler, having three combustion chambers, one to each furnace,	1	14 6	9 7	3	45	6 9	274	3½	9	1640	2000	1582	37·25	22	59·25
Two single-ended boilers, each having two combustion chambers, one to each furnace, . . .	2	11 0	8 4	4	39	5 10	340	3½	5 10	1645	2000	1582	39·2	21·4	60·6

STEEL BOILERS—CONSTRUCTION.

The following is a summary of the Admiralty, Board of Trade, and Lloyd's rules relating to the construction of steel boilers:—

Admiralty tests of Boiler Steel.—All steel to be made by the acid open-hearth process. Every plate, &c., used is to be tested, and must comply with the requirements stated below.

Table CII.—Admiralty Tensile Tests.

Description of Material.	Minimum ultimate tensile strength, tons per square inch.	Maximum ultimate tensile strength, tons per square inch.	Minimum elongation in 8 inches, per cent.
Not exposed to flame & not flanged,	27	30	20%
Exposed to flame and flanged, .	24	27	25%
Rivet bars, ...	24	27	25%
Steam-pipe plates, . . .	24	27	33%
Corrugated or ribbed furnace, .	23	25	27%
Tube forgings (annealed), . .	21	24	23%
Pieces cut from tubes (annealed), .	...	26	27%

For bending tests the specimens are to be heated to a low cherry red, and then cooled in water at 82° F. Strips of plate 1½ inches wide must bend double in press,—inner radius being 1½ times thickness of plate. For pieces of rivet bar, inner radius to equal radius of bar; and for strips from tubes, ½ inch. Plates exposed to flame are also to be tested by welding and forging, some of the welds being broken in the testing machine to ascertain degree of perfection. Angle, tee, and bar steel is to stand such other forge tests as the overseer may direct.

Samples from each batch of rivets are also to stand the following tests:—To be bent double (cold), inner radius of bend being equal to radius of rivet; to be bent double (hot) and hammered till the two parts of shank meet; head to be flattened (hot) without cracking until its diameter is 2½ times diameter of shank; and shank to be nicked on one side, and bent over to show quality of metal.

Tubes under $\frac{5}{16}$ inch thick to stand drifting out at each end, when hot, to 20 per cent. larger diameter, and expanding out, when cold, to 12½ per cent. larger diameter, and thicker tubes to go half these amounts. Sections of thin tubes, 2 inches long, to bear hammering down, cold, to 1¼ inch if lap welded, and to 1 inch if solid drawn; and to bear flattening, if below $\frac{5}{16}$-inch thick, until sides touch, and if thicker until sides come within twice thickness, in all cases without fracture.

Board of Trade Tests of Boiler and other Steel. Instructions to Surveyors.

Manufacture and Testing of Steel Material intended for Boilers and Machinery under Board of Trade Survey.

1. *Introduction.*—The Board having had under consideration the reports issued by the Engineering Standards Committee, prepared the following amended instructions for the guidance of their Surveyors, which came in force on 1st August 1908.

General Conditions.

109. *Process of manufacture: annealing shell plates.* — All steel intended for use in the construction of boilers and for forgings should be made by the open-hearth process. Boiler plates should be of acid quality; but the other portions of boilers, and forgings, may be made of either acid or basic steel. In the case of castings, the steel may be made by any process which has been approved by the Board of Trade.

It is very desirable that all plates (especially those of great thickness) intended for the shells of boilers should be annealed, but it is important that the process should be carefully effected, the plates being heated singly to a suitable temperature, in a properly constructed furnace, and allowed to cool separately and uniformly out of the furnace.

110. *Selection and treatment of test-pieces.* — All the test-pieces required should be selected by the Surveyor, and, except where otherwise specified, the tests should be made in his presence at the place of manufacture, and before the despatch of the material; and the stamping of test-pieces shall be done after all the heating or annealing is completed.

If any material is annealed or otherwise heat-treated, the test-pieces should be similarly and simultaneously treated with the material before they are tested. The specimens should not be further heated, excepting those for temper-bending tests, which should be heated to a blood red and quenched in water at a temperature not exceeding 80° F.

As regards forgings, the test-pieces should be taken from a part of the forgings of sectional dimensions not less than those of the body of the forging, and they should be machined to size without further forging down.

Test-pieces should not be cut off forgings or castings until they have been stamped by the Surveyor after the annealing has been completed.

When a number of articles are cut from one plate, bar or forging, the number of tests required should be the same as that required from the original piece, provided the articles have not been further heated or forged, and can be identified as having formed part of the original piece.

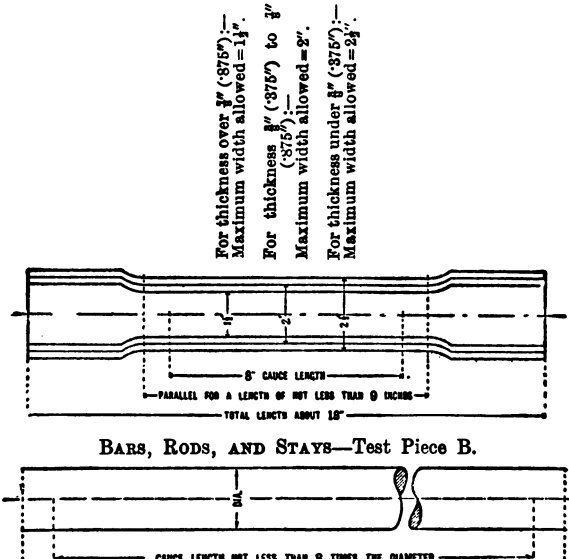
When a number of small forgings are made from the same ingot, or a number of small castings from the same charge of steel, the full

number of tests specified herein need not be made; tensile and bending tests at the rate of one of each for every four articles will, as a rule, in such cases be sufficient.

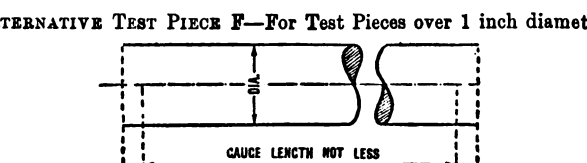
111. *Standard test-pieces.*—The forms and dimensions of test-pieces should be as follows:—

(a) Tensile Tests.

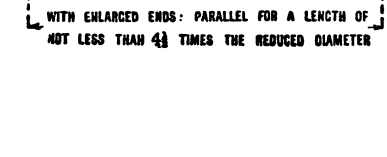
PLATES, TEE, AND ANGLE BARS—Test Piece A.


For thickness over ¾" (·875"):—
Maximum width allowed = 1½".
For thickness ⅜" (·375") to ¾" (·875"):—
Maximum width allowed = 2".
For thickness under ⅜" (·375"):—
Maximum width allowed = 2½".
1½"
2"
2½"
8" GAUGE LENGTH
PARALLEL FOR A LENGTH OF NOT LESS THAN 9 INCHES
TOTAL LENGTH ABOUT 18"

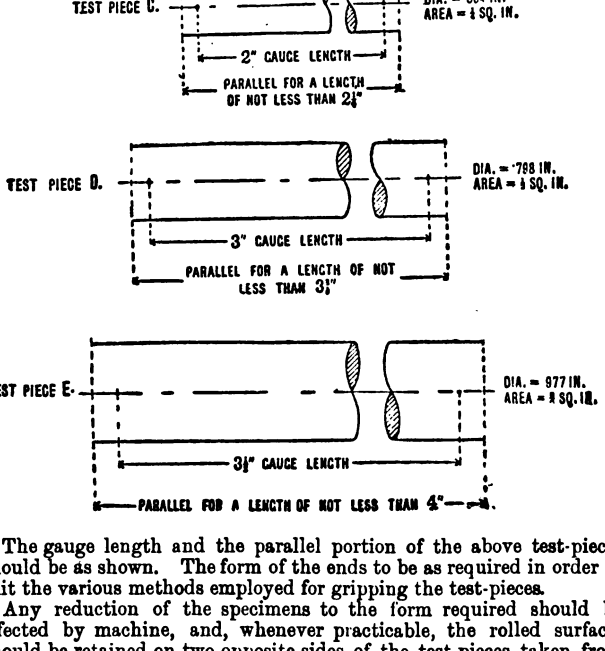
BARS, RODS, AND STAYS—Test Piece B.


DIA.
GAUGE LENGTH NOT LESS THAN 8 TIMES THE DIAMETER
WITH ENLARGED ENDS:—PARALLEL FOR A LENGTH OF NOT LESS THAN 9 TIMES THE REDUCED DIAMETER

ALTERNATIVE TEST PIECE F—For Test Pieces over 1 inch diameter.


DIA.
GAUGE LENGTH NOT LESS THAN 4 TIMES THE DIAMETER
WITH ENLARGED ENDS: PARALLEL FOR A LENGTH OF NOT LESS THAN 4½ TIMES THE REDUCED DIAMETER

FORGINGS AND CASTINGS—Test Pieces C, D and E.


TEST PIECE C.
DIA. = ·564 IN.
AREA = ¼ SQ. IN.
2" CAUCE LENGTH
PARALLEL FOR A LENGTH OF NOT LESS THAN 2½"
TEST PIECE D.
DIA. = ·798 IN.
AREA = ½ SQ. IN.
3" CAUCE LENGTH
PARALLEL FOR A LENGTH OF NOT LESS THAN 3½"
TEST PIECE E.
DIA. = ·977 IN.
AREA = ¾ SQ. IN.
3½" CAUCE LENGTH
PARALLEL FOR A LENGTH OF NOT LESS THAN 4"

The gauge length and the parallel portion of the above test-pieces should be as shown. The form of the ends to be as required in order to suit the various methods employed for gripping the test-pieces.

Any reduction of the specimens to the form required should be effected by machine, and, whenever practicable, the rolled surfaces should be retained on two opposite sides of the test-pieces taken from plates, angle bars, and tee bars.

(b) Bending Tests.

The specimens sheared from plates, angle bars, and tee bars, for bending tests, shall not be less than 1½ inches wide; but for small bars the whole section may be used. The rough edge caused by shearing samples ½ inch in thickness, and above, may be removed by filing or grinding; and samples, 1 inch in thickness, and above, may have the edges machined. The bending specimens of round bars should, whenever practicable, be of the full diameter of the bars as rolled, but those of large section may be turned down to a diameter of 2 inches, if desired.

The bending tests of forgings and castings should be made with

rectangular test-pieces, 1 inch wide by $\frac{3}{4}$ inch thick, which should be machined to size and have the corners rounded to a radius of $\frac{1}{4}$ inch: they should be bent over their thinner section.

112. *Duplicate tests.*—Should either a tensile or a bend test fail to fulfil the test requirements, and the Surveyor considers that the test-piece does not fairly represent the quality of the material, two duplicate specimens may, if the maker wishes, be tested; and if the results obtained from both are satisfactory, the quality of the article shall be judged therefrom and not from the original test which failed. If, however, either of the duplicate tests fails, the article or articles represented shall be rejected.

Should any tensile test-piece break at a point outside the middle half of its gauge length, the test may, at the makers' option and with the Surveyor's approval, be discarded, and another test may be made from the same plate, bar, forging, or casting.

113. *Stamping of material.*—Every article shall be stamped with a number or identification mark such that the charge of steel from which it was made can be readily identified. In addition to this, plates and bars shall be stamped with the maker's name or trade mark, and plates with the results of any tests which are made from them.

114. *Reporting test results.*—After witnessing the steel tests, the Surveyors should, except in the cases subsequently mentioned, submit the form Surveys 24 recording the results as soon after the tests are made as is convenient, without waiting for the completion of the order; and should be forwarded without loss of time to the Surveyors at the port at which the material is to be used.

115. *Freedom from defects, &c.*—The finished material should be sound and free from cracks, surface flaws, and laminations, and no hammer-dressing, patching, burning, or electric welding is permissible.

Local dressing of shell plates is undesirable, but if in any case the steel-makers wish to adopt this method of removing surface defects, the attention of the Surveyor must first be called to the defect, and no dressing must be done until he has inspected the part and satisfied himself that no crack exists. He should then stamp the adjacent part with his initials and record what he has done on the form Surveys 24, which should be forwarded to the Engineer Surveyor-in-Chief for transmission to the Surveyor who will inspect the boiler for which the plate is intended. In no circumstances whatever is hammer-dressing allowed, and the means for removing a surface defect must be confined to chipping and filing.

When the Surveyor who inspects the boiler during construction receives the form Surveys 24, he should, if practicable, arrange for the part to which attention has been drawn to be kept on the outside of the boiler, and he must specially examine the part after it has been rolled to the cylindrical form.

It is most necessary that the Surveyor should carefully scrutinise the inner and outer surfaces of all cyclindrical shell plates, with a view to detecting cracks, while the plates are being worked in the boilers. If he has any doubt about a part, a light chipping should

be taken off the surface, in order to see if the chipping divides at a crack.

Boiler-makers should be requested to examine carefully all shell plates in the various stages of working, as they have the best opportunity of discovering defects, and occasionally cracks develop when working the plates in the boiler shop; but such inspection does not in any way relieve the Surveyor from his duty personally to inspect the plates.

Every precaution should be taken by the steel-makers to prevent a defective plate leaving their works, as they are responsible for supplying sound material. That a great deal can be done in this respect by the makers is clear from the fact that there are some steel-makers manufacturing a large amount of plating with regard to which the Board have no record of a defective shell plate.

In the event of any material proving unsatisfactory in the course of working or machining, it should be rejected notwithstanding any previous certification of satisfactory testing.

Steel for Use in Boilers.

116. *General instructions.*—The following instructions regarding boiler material refer to steel of ordinary mild quality. Where high tensile steel is used, the requirements specified by the Board in each case should be adhered to.

Plates.

117. *Number and nature of tests.*—A tensile and a bending test should be taken from each plate, as rolled; but, when the weight of the plate exceeds two and a half tons, a tensile and a bending test should be taken from each end. Bending tests only, however, need be made from plates for which a greater stress than is allowed for iron is not desired.

The plates for man-hole doors, and for compensating rings around the openings for doors, should be tested in the usual manner.

118. *Tensile strength and elongation.*—The tensile strength of plates not intended to be worked in the fire or exposed to flame, for which special limits have not been sanctioned, should be between 28 and 32 tons per square inch: that of other plates, from 26 to 30 tons per square inch. The elongation should not be less than 20 per cent. in a length of 8 inches for material $\frac{3}{4}$ inch in thickness and upwards which is required to have a tensile strength of 28 to 32 tons per square inch, and not less than 23 per cent. if the tensile strength is required to be between 26 and 30 tons per square inch. For material under $\frac{3}{4}$ inch in thickness, the elongation may be reduced; but, for each eighth of an inch of diminution in thickness, the reduction should not be more than 3 per cent. below the elongations mentioned.

119. *Bend tests.*—Bending test-pieces should withstand being bent,

without fracture, until the sides are parallel at a distance apart of not more than three times the thickness of the specimen. The bending tests of plates not intended to be worked in the fire or exposed to flame may be made with strips in the same condition as the plates: those from other plates should be made with strips which have been tempered.

120. *Witnessing of tests by Surveyor.*—It is very desirable that the Surveyor should witness the whole of these tests; but, in the case of plates made from steel manufactured by any of the makers whose names are given in section 108, he need only select and witness tests from one in four of the plates of each thickness, unless the weight of the plate is over two and a half tons, or special limits of strength, or, in the case of shell plates, a minimum tensile strength exceeding 27 tons is required, in which cases the Surveyor should see the tests made from all the plates.

Angle, Rivet, and Stay Bars.

121. *Number and nature of tests.*—One tensile test should be made from each 15, or part of 15, bars rolled of each section or diameter from the same charge, but not less than two tensile tests should be made, unless the total number of bars rolled from the same charge is 8, or less than 8, and the bars are of the same section or diameter, when one test will suffice. For round bars $1\frac{3}{4}$ inches in diameter, and under, the numbers 50 and 20 should be substituted for 15 and 8 respectively, as determining the number of tests necessary.

A cold and a temper bend should be made from stay bars in the same proportion as that in which tensile tests are required; and a cold or temper bend should be made from each angle or tee bar rolled. No bending tests need be made from rivet bars.

122. *Tensile strength and elongation of stays, angles, and tee bars.*—The tensile strength of longitudinal stays, angles, and tee bars should be between 27 and 32 tons per square inch, with an elongation of not less than 20 per cent. measured on the appropriate standard test-piece (A or B). For bars for combustion-chamber stays, the tensile strength should be between 26 and 32 tons per square inch, with an elongation of not less than 23 per cent. measured on the standard test-piece. When, however, stay bars are tested on a gauge length of four times the diameter (test-piece F), the elongations should be 24 per cent. and 28 per cent. respectively.

For tee or angle bars under $\frac{3}{8}$ inch in thickness, the elongation may be 3 per cent. below that specified for plates.

123. *Bend tests.*—Bending test-pieces should withstand being bent, without fracture, until the sides are parallel at a distance apart of not more than three times the thickness or diameter of the specimen.

124. *Rivet bars.*—The tensile strength of rivet bars should be between 26 and 30 tons per square inch, with an elongation of not less than 25 per cent. measured on the standard test-piece B, or 30 per cent. if measured on test-piece F.

Rivets.

125. *Nature of tests.*—A few rivets of each size should be selected by the Surveyor from the bulk, and should be subjected to the following tests:—

(*a*) The rivet shanks to be bent cold and hammered until the two parts of the shank touch, without fracture on the outside of the bend.

(*b*) The rivet heads to be flattened, while hot, until their diameter is two and a half times the diameter of the shank, without cracking at the edges.

A few check tensile tests of rivets should also be made when the Surveyor considers it necessary. The elongation should, when practicable, be taken in a length of two and a half times the diameter of the prepared part; the tensile strength should be from 27 to 32 tons per square inch and the contraction of area about 60 per cent.

STEEL FORGINGS.

131. *General.*—The forgings shall be made from sound ingots, and not more than the lower two-thirds of the ingot may be utilised for forging. The sectional area of the body of the forging may not exceed one-fifth of the sectional area of the original ingot; and no part of the forging shall have more than two-thirds of the sectional area of the ingot. All ingot steel forgings shall, after completion, be thoroughly annealed at a uniform temperature; and if any subsequent heating is done, the forging shall, if required by the Surveyor, be again annealed.

128. *Number of tests.*—At least one tensile and one bend test shall be taken from each forging; but if the weight exceeds three tons, a tensile and a bending test shall be taken from each end.

129. *Tensile strength and elongation.*—The tensile strength of steel forgings shall not exceed 40 tons per square inch; and the elongation, measured on the appropriate standard test-piece C, D, or E, shall not be less than 17 per cent. for 40-ton steel; and in no case may the sum of the tensile strength and the corresponding elongation be less than 57.

130. *Bend tests.*—The bending test-pieces should withstand being bent through an angle of 180° without fracture; the internal radius of the bend being not greater than that specified below:—

Maximum specified tensile strength of forging.	Internal radius of test-piece after bending.
	inch.
Up to 32 tons per square inch	$\frac{1}{2}$
Above 32 tons and up to 36 tons per square inch	$\frac{5}{8}$
,, 36 ,, 40 ,, ,,	$\frac{3}{4}$

Steel Castings.

135. *General.*—All steel castings shall be thoroughly annealed at a uniform temperature and shall be allowed to cool down prior to removal from the annealing furnace; and if subsequently heated, with the Surveyor's approval, shall again be similarly annealed, if required by the Surveyor.

132. *Number and nature of tests.*—No tests need be made from unimportant steel castings or from steel castings which are used for articles usually made of cast-iron, if the scantlings are not materially reduced below what would be required if cast iron were used. All other steel castings shall be tested as follows:—

At least one tensile and one bending test should be made from the castings from each charge; and where a casting is made from more than one charge, at least four tensile and four bending tests should be made from pieces cast as far apart as possible on the casting and as near the top and the bottom respectively as practicable.

When more than one casting is made from one charge, at least one tensile and one bending test should be made from the castings run from one common pouring head; but separate tests should be made from each casting or set of castings run from each separate pouring head. Small castings may, however, be dealt with in accordance with the provisions of section 110.

133. *Tensile strength and elongation.*—The tension strength may range from 26 to 40 tons per square inch, with an elongation, measured on the standard test-piece C, D, or E, of not less than 15 per cent. If, however, the castings are to be used for the more important pieces of machinery, such as pistons, etc., or for articles usually made of wrought material, the elongation should not be less than 20 per cent. where the corresponding tensile strength is between 26 and 35 tons per square inch.

134. *Bend tests.*—The bending test-pieces must withstand being bent, without fracture, through an angle of 60° if the tensile strength is between 35 and 40 tons per square inch; and in the case of other castings, through an angle of 90°. But if they are required to be of the superior quality referred to above, the angle must not be less than 120°.

Tubes.

126. *Number and Nature of Tests.*—

(a) *Solid-drawn Steel Steam Pipes, Boiler Tubes, &c., subject to Internal Pressure.*

The makers should take a few samples from each batch of tubes and test them for tensile strength and elongation. A bending test should also be made by them from the scrap end of each tube drawn.

Tensile and bending tests should also be made in the Surveyor's

presence from specimens selected by him in the following proportion, from the tubes made from *each charge*:—

Tubes up to and including 3 inches in diameter : 1 in 40 or part thereof.

Tubes above 3 inches up to and including 4 inches in diameter : 1 in 20 or part thereof.

Tubes above 4 inches up to and including 5 inches in diameter : 1 in 10 or part thereof.

Tubes above 5 inches up to and including 7 inches in diameter : 1 in 6 or part thereof.

Tubes above 7 inches in diameter : 1 in 4 or part thereof.

The tensile strength should not exceed 28 tons per square inch, with a minimum elongation of 25 per cent. in a length of 8 inches, or 23 per cent. if the thickness of the tubes is less than $\frac{1}{4}$ of an inch.

All the tubes should be tested by the makers to a suitable hydraulic pressure, and the tests of at least 25 per cent. of them should be witnessed by the Surveyor. The tests of all steam pipes should, however, be witnessed by the Surveyor on completion of the pipes, that is, after they have been bent to shape and the flanges have been secured in position (*see* section 174).

(b) *Solid-drawn Steel Tubes subject to External Pressure.*

If no allowance over that given for iron tubes is required, a few bending tests should be made from the scrap ends of the stay tubes, but special tests need not be made from the ordinary tubes if the Surveyor finds the general quality of the material satisfactory and he is satisfied.

If allowance over iron is required, tensile and bending tests should be witnessed by the Surveyor in the proportions given for solid-drawn steel steam pipes. The tensile strength should range between 23 and 30 tons per square inch, and the elongation should be at least that required for similar solid-drawn steam pipes.

All the tubes should be tested by the makers to a suitable hydraulic pressure, but the tests need not be witnessed by the Surveyor if he is satisfied that the tubes have been duly tested by the makers.

(c) *Steel Lap-welded Tubes subject to External Pressure.*

(i) *Steel tubes for which no allowance over iron is required.*—A few bending tests should be made from the scrap ends of the stay tubes or the strips from which they are made, but special tests need not be made from the ordinary tubes if the general nature of the material has been found satisfactory and the Surveyor is satisfied.

(ii) *Steel stay tubes for which allowance over iron is required.*—Tensile and bending tests should be made from 25 per cent. of the *strips* from which the tubes are made. The tensile strength should not exceed 28 tons per square inch, and the elongation should be at

least 25 per cent. in a length of 8 inches when the strips are tested in their normal condition.

All the tubes should be tested by the makers to a suitable hydraulic pressure, but the tests need not be witnessed by the Surveyor if he is satisfied that the tubes have been duly tested by the makers.

127. *General Requirements for all Tubes.* — The hydraulic test should not, in any case, be less than three times the working pressure, and it should not exceed four times the pressure given by the rule:—

$$\frac{6000 \times \text{thickness in inches}}{\text{inside diameter in inches}} = \text{pressure},$$

in the case of lap-welded tubes, or five times that pressure in the case of solid-drawn steel tubes.

All the tests mentioned should be made in the Surveyor's presence, except where otherwise stated, and such means as may be necessary should be taken to satisfy the Surveyor that the specimens he may have to test have been cut from the tubes they represent.

If any of the aforesaid tubes are made in long lengths and passed by the Surveyor in that condition, the number of tests required may be calculated on the number of tubes as made, notwithstanding that they may afterwards be cut up into shorter lengths.

Table CIII.—Board of Trade Tensile Tests.

Description of Material.	Minimum ultimate tensile strength, tons per square inch.	Maximum ultimate tensile strength, tons per square inch.	Elongation on 10 inches, per cent.
Plates not exposed to flame, . . .	28	32	20% in 8 ins.
Plates that are exposed to flame, .	26	30	Not less than 23% for annealed plates.
Rivet bars, . .	26	30	Not less than 25%.
Stay bars, ordinary .	28	32	Not less than 20%.
Stay bars, combustion chamber . .	26	30	Not less than 23%.
Tube strips . .	26	30	About 25%. Not less than 20%.
Rivets, . . .	28	32	Contraction of area about 60%.

Shell plates, special, on application up to 36 tons.

Steel tubes should be made of open-hearth acid steel, unless material of other quality has been specially approved for the purpose. Solid-drawn tubes of a thickness exceeding $\frac{1}{8}$ inch should be finished by the hot-drawn process, unless cold-drawing has been specially sanctioned, and all cold-drawn tubes should afterwards be efficiently annealed.

STEEL BOILERS.

136. *Thickness of plates: drilling, welding, and annealing.*— The thickness of plates, other than tube strips, used in the construction of boilers should not be less than $\frac{5}{16}$ inch.

It is expected that the rivet holes will be drilled, and not punched. Plates that are drilled in place should be taken apart and the burr taken off, and the holes slightly countersunk from the outside.

Butt straps should be cut from plates, and not from bars.

Steel plates which have been welded should not be passed if subject to a tensile stress, and those welded and subject to a compressive stress should be efficiently annealed.

Local heating of the plates should be avoided, as many plates have failed from having been so treated.

All plates that have been flanged or locally heated, and all stays and stay tubes which have been locally heated, should be carefully annealed after being so treated.

137. *Cylindrical Boiler Shells* shall be of steel made by the Open Hearth Process, acid or basic. Generally it shall be of the 28/32 tons ultimate tensile standard quality, tested in accordance with British Engineering Standards Rules. Steel of a higher ultimate tensile may be used if desired, by arrangement, the tests also to be standard ones. The elongation must be not less than 20 per cent. in 8 inches. Whatever steel is ordered there must be a 4-ton range allowed to the makers, unless otherwise agreed to.

All the holes for rivets must be drilled in place as far as possible, the burrs removed, and the faying surfaces cleaned, and the edges of holes eased, then the following holds:—

Rule . Working Pressure $= \frac{(t-2) \times S \times J}{C \times D}$.

S is the minimum tensile strength of the shell plates in tons.
J is the percentage of strength of the longitudinal seams.
D is the inside diameter of the outer strake of plating in inches.
C when the longitudinal seams have double butt straps 2·75.
C ,, ,, lap joints and are treble riveted 2·83, when double riveted 2·9, and when single riveted 3·3.

The riveting of the seams joining the end plates to the cylindrical shell shall be not less than 42 per cent. of that of the shell plate.

When the shell plates are over $^{29}/_{32}$nds inch thick there shall be double riveting.

The circumferential seam at or near the middle of the single-ended boiler shall have a strength of joint not less than 60 per cent. of the solid plate. The inner seams of double-ended boilers shall have 62 per cent. When single-ended boilers are over $^{44}/_{32}$nds inch and double-ended $^{38}/_{32}$nds inch, there shall be three rows of rivets in these middle seams. When the shell plates exceed $^{16}/_{32}$nds inch thick these seams of double-ended boilers shall be double-riveted.

RULES FOR STRENGTHS OF PARTS OF BOILERS.

Board of Trade rules for cylindrical shells.—In all calculations for strength of shells the *minimum* strength of plate, as disclosed by the tests, must be used.

To ascertain the strength of shell, the relative sectional areas of plate and rivet must first be determined by the following formulæ :—

Rule 248. Percentage of strength of pitch plate :—

(i.) At joint compared with solid plate $= \dfrac{100(p-d)}{p}$.

(ii.) Percentage of strength of rivets as compared with solid plate $= 100\,\dfrac{(S_2 \times a \times n \times C)}{S_1 \times p \times T}$.

(iii.) Percentage of combined strength of plate at inner row of holes and of rivets in the outer row $= \dfrac{100(p-2d)}{p} + \dfrac{100(S_2 \times a \times C)}{S_1 \times p \times T}$.

Where p = pitch of rivets in outer rows in inches.
d = diameter of rivets in outer rows in inches.
a = sectional area of one rivet.
n = number of rivets fitted in the pitch, p.
T = thickness of plate in inches.
C = 1·0 for rivets in single shear.
C = 1·875 for rivets in double shear.
S_1 = the minimum tensile strength of plate in tons.
S_2 = the shearing strength of rivets taken usually as 23 tons.

Thickness of Butt Straps.—The outer one should be in thickness 0·625 that of the plate; in every case it must be thick enough to withstand caulking. The inner strap should be $^{4}/_{32}$nds inch thicker than this.

When the rivets in the inner rows of the seam are double that of the outer, as is generally the case with longitudinal joints, then for the outer strap

$$t = \frac{5 \times (p-d)}{8 \times (p-2d)} \times \mathrm{T}.$$

For the inner strap add $^{4}/_{32}$nds inch to the above.

Rule 249. Diameter of rivet $= \frac{t}{1{\cdot}21 \times n} \times \frac{x}{1-x}$.

Pitch of rivet = diameter $\div (1-x)$.

n = number of rivets per pitch of outer rows.

x = the strength of joint as a fraction of the pitch $= \frac{p-d}{p}$.

d = diameter of rivet, and p the pitch in outer rows in inches.

t = the thickness in 32nds of an inch.

Rules for riveting longitudinal seams when butted and fitted with double straps; the strength of the rivets against shear at 23 tons and the plate for tensile 27 tons (*vide* Table CIV):—

n = the number of rivets per pitch of outer rows.
x = the strength of joint as a fraction of the pitch.
d = the diameter of rivet.
p = pitch of outer rows.

Rule 249a. Diameter of rivet $= \frac{t}{1{\cdot}255n} \times \frac{x}{1-x}$.

Rule 249b. Pitch of rivet $= \frac{\text{diameter}}{1-x}$.

These are deduced from the following equations:—

(*a*) $$n \times 7854d^2 \times 1{\cdot}875 \times 23 = (p-d)27t,$$

that is, $$1{\cdot}255nd^2 = (p-d)t.$$

(*b*) $$\frac{p-d}{p} = x.$$

N.B.—The Board of Trade require the rivets diameter in all cases to be not less than the thickness of plate.

139. *Openings in shells; doors, etc.*—The openings in the shells of cylindrical boilers should have their shorter axes placed longitudinally.

Compensating rings of at least the same effective sectional area as the plates cut out, and not less in thickness than the plates to which they are attached, should be fitted around all man-holes and openings, or the surrounding portion of the plates otherwise efficiently stiffened.

It is very desirable that the compensating rings around openings in flat surfaces be made of L- or T-bars. When a ring is not fitted around such an opening, and the plate is flanged for compensation, the total depth, D, of the flange should not be less than that given by the following equation:—

$$D = \sqrt{\text{width of opening} \times \text{thickness of plate}}.$$

Table CIV.—Relative Thickness of Boiler Shell Plates for Different Tensile Strengths.

Minimum tensile test strength in tons per square inch.											
27	28	29	30	31	32	33	34	35	36	37	38
0·50	0·483	0·466	0·450	0·486	0·422	0·409	0·397	0·386	0·375	0·365	0·355
0·60	0·579	0·559	0·540	0·524	0·507	0·491	0·476	0·463	0·450	0·438	0·427
0·70	0·675	0·652	0·630	0·611	0·591	0·573	0·551	0·540	0·525	0·511	0·498
0·80	0·772	0·745	0·720	0·699	0·675	0·655	0·635	0·617	0·600	0·584	0·569
0·90	0·868	0·838	0·810	0·784	0·760	0·736	0·714	0·694	0·675	0·657	0·640
1·00	0·965	0·931	0·900	0·873	0·844	0·818	0·793	0·771	0·750	0·730	0·711
1·10	1·061	1·024	0·990	0·961	0·929	0·900	0·872	0·848	0·825	0·803	0·782
1·20	1·157	1·117	1·080	1·048	1·013	0·982	0·925	0·925	0·900	0·876	0·853
1·30	1·254	1·211	1·170	1·135	1·097	1·064	1·031	1·003	0·975	0·949	0·924
1·40	1·350	1·304	1·260	1·223	1·182	1·145	1·110	1·080	1·050	1·022	0·996
1·50	1·447	1·397	1·350	1·310	1·266	1·227	1·190	1·157	1·125	1·095	1·067
1·60	1·543	1·490	1·440	1·397	1·351	1·309	1·269	1·234	1·200	1·168	1·138
1·70	1·640	1·583	1·530	1·484	1·435	1·391	1·348	1·311	1·275	1·241	1·209
1·80	1·736	1·676	1·620	1·572	1·519	1·473	1·428	1·388	1·350	1·314	1·280
1·90	1·833	1·769	1·710	1·659	1·604	1·554	1·507	1·465	1·425	1·387	1·350
2·00	1·929	1·862	1·800	1·746	1·688	1·636	1·586	1·542	1·500	1·460	1·421

When chain riveted, the distance between the outer and the next row of rivets should be not less than $0{\cdot}33p + 0{\cdot}67d$ or $2d$, whichever is greater. When there are the full number of rivets in contiguous rows, the distance apart must be not less than $2d$. When zig-zag riveted, these distances are $0{\cdot}2p + 1{\cdot}15d$ and $0{\cdot}2p + 0{\cdot}4d$. In each case p is the pitch in the outer rows.

The maximum pitch of rivets in longitudinal seams $= C \times T + 1\frac{5}{8}$ inch. C is a coefficient as follows, and T is the thickness of plate or strap in inches.

Number of Rivets per Pitch.	Coefficient C for Lap Joints.	Coefficient C for Butt Joints.
1	1·31	1·75
2	2·62	3·50
3	3·47	4·63
4	4·14	5·52
5	...	6·00

N. B.—The Board of Trade present limit of pitch is 12·25 ins.

BOARD OF TRADE AND REGISTRATION SOCIETIES' RULES FOR AUXILIARY AND SPECIAL BOILERS.

An auxiliary boiler is one used generally for purposes other than that of supplying the main propelling machinery, and while it may be so used on an emergency, it is not a necessary part of the main boiler installations. It is, however, supplied with feed water from a surface condenser or with other equally pure water except on emergency.

Winch (donkey) boilers are those having no connection with the main boilers but employed on winches, cranes, and other appliances generally outside the engine and boiler rooms, and whose normal supply of feed water is from the sea or other impure water.

All horizontal cylindrical boilers whose diameter exceeds 10 feet must be made in accordance with the Rules for Main Boilers, as must boilers of all sizes which supply steam to main machinery.

Vertical boilers must have their circumferential seams equal to not less than 42 per cent. of the solid plate. The rivets must not exceed 1·5 × thickness of plate, and when the seams are not complete circles and when the plates exceed $^{2}\%_{32}$nds inch in thickness, the riveting shall be double.

In boilers whose normal feed supply is not pure water, the spacings of the smoke tubes must be increased over those given for ordinary boilers by $^{5}/_{32}$nds inch.

Mud holes and sight holes must be provided for cleaning and scaling cross and other tubes, and they must be in accessible places.

Plain Vertical Furnaces.

1. When tapered the diameter for calculation purposes shall be the mean of that at top and that at bottom where it has substantial support from flange or ring. The length shall be from centres of rivets at crown and the row at bottom. When there are rows of screwed stays whose distance is not more than 14 times the thickness of furnace plate when simply riveted over, or 16 times when fitted with nuts, it may be measured to them. Such screwed stays must be in diameter over the threads not less than 2·5 times the thickness of furnace plate.

2. When the furnaces are spherical in form and convex upward at their tops, and are without support from stays of any kind,

$$\text{Working pressure} = \frac{275\ (t-1)}{R}$$

where t is the thickness of the top plate in 32nds of an inch,
R is the outer radius of curvature of the furnace in inches.

3. For the ogee ring which connects the bottom of the furnace to the shell, and sustains the whole load on the furnace vertically,

$$\text{Working pressure} = \frac{145\ (t-1)^2}{D \times (D-d)}$$

where t is the thickness of the ogee ring in 32nds of an inch.
D is the inside diameter of the boiler shell in inches.
d is the outside diameter of the lower part of the furnace where it joins the ogee ring.

Tubes and Tube Plates.

4. When vertical boilers have a nest or nests of horizontal tubes so that there is direct tension on the tube plates due to the vertical load on the boiler ends or to their acting as horizontal ties across the shell, the thickness of the tube plates and the spacing of the tubes must be such that the section of metal taking the load is sufficient to keep the stress within that allowed on the shell plates.

Further, each alternate tube in the outer vertical rows of tubes must be a stay tube. The tube plates between the stay tubes must be in accordance with the rules for tube plates, as in par. 11, Section III., Part I., and in addition

$$\text{Working pressure} = \frac{(t-2) \times S \times (p-d)}{2{\cdot}9 \times D \times p}$$

where S is the minimum tensile strength of the steel plate in tons per square inch.
t is the thickness of the tube plate in 32nds of an inch.
D is twice the radial distance of the centre of the outer row of tube holes from the axis of the shell in inches.
p is the vertical pitch of tubes.
d is the diameter of the tube holes in inches.

The spaces between the tubes of such vertical boilers should not be less than $1\frac{1}{8}$ inches at the back ends.

Tops of Vertical Boilers.

5. When these are dished or spherical in form and without stays they must be in accordance with par. 17, Section III., Part I.

6. When the top is a complete hemisphere and without stays or other supports, and is made in more than one plate,

$$\text{Working pressure} = \frac{(t-2) \times S \times J}{C \times R}$$

where t is the thickness of top plates in 32nds of an inch.
S is the minimum tensile strength of the steel plates in tons per square inch.
J is the strength of riveted joint per cent. of solid plate.
R is the inner radius of curvature in inches.
C for single riveting is 3·3; for double riveting C is 2·9; for treble riveting 2·83.

WATER TUBE BOILERS.

These boilers should be constructed generally in accordance with the rules for cylindrical boilers, so far as they are applicable. Their cylindrical steam receivers and water chambers are usually of comparatively small diameter, and inasmuch as there may be considerable variations in pressure in these boilers occurring with a rapidity hardly possible with tank boilers from the small amount of water contained, and that often in a state of violent ebullition, it is requisite that internally they shall be practically perfectly cylindrical. They should be made without lapped joints longitudinally. They should by preference be welded with an external cover strap, or, when possible, be solid drawn. If the latter, the thick tube plate necessary for tube holes to be parallel cannot be obtained. When the holes are radial, a thinner tube plate will do, and the water drums may be solid drawn.

The tubes must be solid drawn, and by preference cold finished, so as to have a smooth surface and uniformity of thickness. They must be made from steel produced by an open hearth process, acid or basic, and having an ultimate tensile strength not exceeding 27 tons, with an elongation not less than 25 per cent. in 8 inches.

These tubes should be carefully annealed, after which they should be capable of being flattened until the inner surfaces are within a distance apart at most equal to twice the thickness, and their ends must withstand expansion by rollers by 12·5 per cent. when not over 10 L.S.G. thick, 9·5 up to 6 L.S.G.; over that thickness by 6·5, without signs of cracking. Every tube must be tested by hydraulic pressure to $4 \times$ W.P., and none less than 1000 lbs. per square inch.

$$\text{The thickness of the tubes} = \frac{\text{W.P.} \times d}{\text{F}} + 8 \text{ in 100ths of an inch.}$$

d is the external diameter in inches.

F is 60 for the two rows next the fire, and in the gaps in the nests of tubes through which the hot gases outflow. For the remainder F = 75.

The tube plates forming portions of the drums must have a thickness in line with the tube holes as follows :—

$$\text{Thickness in 32nds of an inch} = \frac{\text{W.P.} \times \text{D} \times p}{867(p - d)} + 4.$$

D is the inside diameter of drums in inches.

d is the external diameter of tubes in the line in inches.

p is the pitch of tubes in the line parallel with drum axis in inches.

The minimum strength of plate is assumed to be 26 tons.

In the case of headers as in the Babcock and similar boilers,

$$\text{Thickness in 32nds of an inch} = b \times \sqrt{\frac{\text{W.P.}}{\text{C}}} + 4.$$

When the surface is unpierced with holes

b is the breadth in inches of their flat surface from support to support.

$C = 80$ for wrought steel or 60 for steel castings.

When headers are staggered or formed in such a way as to resist deformation, they should be $1\frac{2}{32}$nds inch thick, and in no part or patch less than $\frac{8}{32}$nds inch.

The thickness at the tube holes should be in 32nds inch, $4 \times \sqrt{\text{diameter of tube in inches}} + 8$.

Safety valves.—Aggregate area in inches = total heating surface in feet $+ (p + 15)$.

The feed apparatus should be automatic in action, simple and effective. There should be two independent means of admitting feed water placed in such a position as not to check the circulation.

The tubes should have a neck bearing in the plate holes not less than half an inch and project through at least a quarter of an inch and be "bell-mouthed," so that the diameter of end is $(d + 2)$ thirty-seconds of an inch in excess of d of that of the tube hole.

CAST METAL PIPES AND THEIR EQUIVALENTS.

Recommendations of the British Marine Engineering Design and Construction Committee.

(1) **Cast steel** should be used instead of cast iron when the temperature exceeds 425° F., which should have a minimum tensile strength of 28 and a maximum of 35 tons, with an elongation of 15 per cent. in 2 inches. When the castings are intricate, a higher tensile may be permitted with an elongation of 12·5 per cent.

They should be tested by water up to 3 times the working pressure they may have to bear.

When subject to sea-water the manganese content should not exceed 0·4 per cent.

(2) **Cast iron** for boiler mountings and all fittings and pipes exposed to pressure exceeding 75 lbs. should have a tensile strength of not less than 9 tons, and test-bars 1·5 inches square on supports 6 inches apart should withstand the impact of a 21 lbs. weight dropped 4 times from a height of 15 inches without fracture.

(3) **Bronze** for mountings and fittings exposed to temperatures over 350° F. should be of an alloy of 87·5 Cu, 10 Sn, 2 Zn, and 0·5 Pb, or other equally good and satisfactory metal. Their ultimate tensile strength at a temperature of 550° should be not less than 14 tons, with an extension of 10 per cent. in 2 inches.

(4) **Good commercial bronze** for general purpose should have that tensile, etc., at 350° F.

All feed-delivery fittings should be tested to $2{\cdot}5 \times$ W.P., and others to $2 \times$ W.P.

Their working pressure and thickness shall be determined as follows:—

$$\text{Working pressure} = \frac{(t-x)}{D} \times K$$

$$\text{Thickness } t = \frac{\text{W.P.} \times D}{K} + x.$$

D is the internal diameter in inches.
t is the thickness in 100ths or 32nds of an inch.
K indicates the stress.
x is an added factor for toleration, etc.

Table of Values of K and x.

Material of Casting.	When t is in 32nds.		When t is in 100ths	
	K.	x.	K.	x.
Cast steel (28/35 tons tensile)	400	8	128	25
Cast iron (at least 9 tons tensile)	200	6	64	19
Bronze, Admiralty and equally good	220	4	70	13
Bronze, good mercantile .	175	4	56	13
Bronze, commercial quality unknown	150	4	48	13

The minimum thickness for cast metal pipes and their equivalents shall be as given by the following Rules, where D is the inside diameter in inches, and t the thickness in 32nds of an inch.

(a) When of cast steel $t = 2{\cdot}5\sqrt{D} + 6$.
(b) When of cast iron $t = 2{\cdot}5\sqrt{D} + 4$.
(c) When of cast bronze $t = 2{\cdot}5\sqrt{D} + 2$.

RULES FOR THICKNESS OF FLAT PLATES.

The following are the rules now in use for flat plates supported by stays by the Board of Trade and the Register Societies:—

Rule 252. $\text{Working pressure} = \frac{(t-1)^2 \times C}{a^2 \times b^2}$.

t is the thickness of plate in 32nds of an inch.
t_w the thickness of washers, strips, or doubling plates when so fitted in 32nds of an inch.
a is the distance apart of the stays in the rows in inches.
b is the pitch of the stays in the rows.
C a coefficient depending on the method of fitting.

(*a*) Where plates are exposed to flame, the stays are screwed into the plates and the ends riveted over, C=50; when not exposed to flame, 57. The plate must be in thickness at least half the diameter.

(*b*) Where stay tubes are expanded in their holes, C=52; if fitted with nuts, 72.

(*c*) Where plates are exposed to flame, and the stays are screwed into the plate and fitted with nuts on the outside, C=75; and when not exposed to flame, C=86.

(*d*) When not exposed to flame and stays pass through the plates with nuts outside and inside, C=96.

(*e*) Where plates are stiffened by flanging, C=110 when not exposed to flame, and C=96 when exposed. The pitch to be reckoned from the commencement of the curvature.

(*f*) When the plates are supported by stays passing through them, and are fitted with nuts inside and washers and nuts outside, the diameter of the washer being at least 3·5 × diameter of stay, and the thickness at least two-thirds that of the plate,

$$\text{W.P.} = \frac{100}{a^2 \times b^2}\{(t-1)^2 + 0{\cdot}15t_w^2\}.$$

(*g*) When the washers have a diameter of at least 0·67 the pitch of stays, and at least 0·67 the thickness of plate, and not more than the plate in thickness and riveted to the plate, the added part is

$$0{\cdot}35t_w^2 \text{ instead of } 0{\cdot}15t_w^2.$$

(*h*) When strips of that breadth and thickness are riveted to the plate, the added part is

$$0{\cdot}55t_w^2.$$

(*k*) With full doubling plates riveted on it is $0{\cdot}85t_w^2$.

(*m*) **Back and front tube plates.**—For portions of tube plates in the nests of tubes:—

$$\text{W.P.} = \text{C} \times \frac{(t-1)^2}{p}.$$

t is the thickness of tube plates in 32nds of an inch.

p is the mean pitch of stay tubes, being the sum of the four sides of the quadrilateral divided by 4.

C=38 when the tubes are screwed and expanded in the plates, and when nuts are also fitted, C=49.

The Rules of the B.M.E.D. & C. Committee for flat surfaces are:—

Rule 252a. Working pressure $= (t-1)^2 \times \text{C} \div (a^2 + b^2)$,

t is the thickness in 32nds inch.

t_n that of washers and doublings.

a the distance apart of rows of stays whose pitch is b.

C a coefficient.

C for screwed stays with ends riveted only, 50.
C ,, ,, ,, ,, nuts on outside of plates, 75, and when exposed to flame, 88.
C for stay tubes without nuts, 55 ; with nuts, 72.

When the stays pass through the plates and have nuts inside, and outside nuts with washers of diameter 3·5 the diameter of stay and at least 0·2 × diameter of stay in thickness,

Rule 252b. Working pressure $= \frac{100}{a^2+b^2}\{(t-1)^2+0{\cdot}15t_n^2\}$.

When the washer is 0·66 the pitch and riveted to plate,

Rule 252c. Working pressure $= \frac{100}{a^2+b^2}\{(t-1)^2+0{\cdot}35t_n^2\}$.

When there are doubling straps in width 2/3, pitch and riveted to plate,

Rule 252d. Working pressure $= \frac{100}{a^2+b^2}\{(t-1)^2+0{\cdot}55t_n^2\}$.

Front tube plates.

Rule 252e. Working pressure $= C(t-1)^2 \div p^2$,

p is the mean pitch and C = 38 screwed only, and 49 with nuts.

Wide spaces in front tube plate.

Rule 252f. Working pressure $= C\{(t-1)^2+0{\cdot}55t_n^2\} \div a^2+b^2$,

a is the horizontal pitch of stay tubes across the wide space.
b is the vertical pitch in the bounding rows.
C = 55 when screwed only, and 72 with nuts ; when there are nuts only on alternate rows, C = 63.

Compression on tube plates.

Rule 252g. Working pressure $= \frac{875 \times (D-d) \times t}{B \times D}$,

where D is the horizontal distance apart of the tubes.
d is the internal diameter of tubes.
B the width of combustion chamber.

Table CVI.—Pitch of Stays and Area of Flat Surfaces of Combustion Chambers (Board of Trade, Lloyd's, etc.).

Pressure in lbs. per square inch.	Stays screwed through plates and fitted with nuts at ends.									
	Plates $^{16}/_{32}$ inch.		Plates $^{18}/_{32}$ inch.		Plates $^{20}/_{32}$ inch.		Plates $^{22}/_{32}$ inch.		Plates $^{24}/_{32}$ inch.	
	Pitch.	Surf.	Pitch.	Surf.	Pitch.	Surf.	Pitch.	Surf.	Pitch.	Surf.
150	7·49	56·1	8·50	72·2	9·50	90·2	10·49	110	11·49	132
155	7·37	54·3	8·37	70·0	9·34	87·3	10·30	106	11·31	128
160	7·25	52·6	8·22	67·6	9·20	84·6	10·15	103	11·09	123
165	7·14	51·0	8·11	65·7	9·06	82·0	10·00	100	10·95	120
170	7·04	49·5	7·98	63·7	8·92	79·6	9·86	97·3	10·73	116
175	6·94	48·1	7·87	61·9	8·79	77·3	9·72	94·5	10·63	113
180	6·84	46·8	7·76	60·2	8·67	75·1	9·57	91·6	10·48	110
185	6·75	45·5	7·65	58·5	8·56	73·2	9·46	89·4	10·34	107
190	6·66	44·3	7·55	57·0	8·44	71·2	9·33	87·0	10·20	104
195	6·57	43·2	7·45	55·5	8·33	69·4	9·21	84·9	10·05	101
200	6·49	42·1	7·36	54·2	8·23	67·7	9·09	82·7	9·95	99·1
205	6·41	41·1	7·27	52·8	8·13	66·0	8·98	80·6	9·83	96·6
210	6·39	40·9	7·19	51·6	8·03	64·4	8·87	78·7	9·72	94·4
215	6·33	40·0	7·10	50·4	7·93	62·9	8·77	76·9	9·60	92·2
220	...	...	7·01	49·1	7·84	61·5	8·67	75·2	9·49	90·1
225	...	...	6·94	48·1	7·76	60·2	8·57	73·5	9·39	88·1
230	...	...	...	...	7·67	58·8	8·48	71·9	9·28	86·2
235	...	...	...	...	7·59	57·6	8·39	70·3	9·19	84·4
240	...	...	...	...	...	...	8·30	68·9	9·09	82·6
245	...	...	...	...	...	...	8·20	67·3	9·00	81·0
250	...	...	...	...	...	...	8·13	66·1	8·91	79·3

N.B.—When plates are not exposed to flame, multiply the *surfaces* above by 1·173, and the pitches by 1·083.

Table CVII.—Pitch of Stays supporting Flat Plates not exposed to Flame by Board of Trade, Lloyd's, etc. Rule. (Thicknesses in 32nds of an inch.)

Working Pressure, lbs.	Stays are fitted with nuts inside and out. Washer under outer is 3·5 diameter of stay.									
	Plate 20. Washer 14.	Plate 22. Washer 15.	Plate 24. Washer 16.	Plate 26. Washer 17.	Plate 28. Washer 19.	Plate 30. Washer 20.	Plate 32. Washer 22.	Plate 34. Washer 22.	Plate 36. Washer 22.	Plate 38. Washer 22.	Plate 40. Washer 22.
150	11·40	12·58	13·75	14·92	16·16	17·42	18·46	19·73	20·80	...	...
155	11·20	12·38	13·52	14·68	15·88	17·08	18·22	19·41	20·47	...	...
160	11·04	12·18	13·31	14·45	15·64	16·77	17·98	19·11	20·14	21·23	...
165	10·04	12·00	13·11	14·23	15·33	16·52	17·70	18·82	19·90	20·92	...
170	10·71	11·82	12·91	14·02	15·17	16·28	17·44	18·54	19·54	20·61	21·65
175	10·56	11·65	12·73	14·81	14·95	16·23	17·19	18·27	19·26	20·31	21·34
180	10·41	11·49	12·58	13·62	14·75	15·81	16·95	18·01	18·98	20·02	21·04
185	10·27	11·33	12·40	13·45	14·55	15·60	16·72	17·76	18·73	19·73	20·78
190	10·13	11·18	12·21	13·29	14·37	15·40	16·50	17·53	18·49	19·44	20·48
195	10·00	11·04	12·05	13·15	14·18	15·20	16·28	17·31	18·25	19·21	20·22
200	9·87	10·90	11·91	12·92	14·00	15·01	16·08	17·09	18·02	18·99	19·96
205	9·75	10·77	11·76	12·76	13·82	14·83	15·88	16·83	17·80	18·76	19·71
210	9·63	10·64	11·62	12·61	13·65	14·65	15·69	16·68	17·58	18·53	19·46
215	...	10·51	11·48	12·46	13·49	14·47	15·51	16·49	17·37	18·32	19·25
220	...	...	11·35	12·32	13·34	14·30	15·33	16·30	17·17	18·11	19·04
225	...	...	...	12·18	13·19	14·15	15·16	16·12	16·99	17·91	18·83
230	...	...	...	12·05	13·05	14·00	14·99	15·94	16·81	17·72	18·62
235	...	...	...	...	12·91	13·85	14·83	15·77	16·63	17·52	18·42
240	...	...	...	...	12·77	13·71	14·68	15·60	16·45	17·33	18·22
245	...	...	...	...	...	13·57	14·53	15·44	16·28	17·16	18·04
250	...	...	...	...	...	13·43	14·39	15·28	16·12	17·00	17·86

N.B.—For surface pitch$^2 = \frac{a^2 + b^2}{2}$.

Table CVIII.—Pitch of Stays supporting Flat Plates when fitted with Washers 0·66 of the Pitch in Diameter Riveted to them outside. By Board of Trade, Lloyd's, etc. Rule.

Working Pressure, lbs.	Thicknesses of plates and washers in 32nds of an inch.													
	Plate 20.		Plate 22.		Plate 24.		Plate 26.		Plate 28.		Plate 30.		Plate 32.	
	Washer		Washer		Washer		Washer		Washer		Washer		Washer	
	14.	20.	16.	22.	18.	24.	20.	26.	21.	28.	22.	30.	23.	32.
150	11·97	12·92	13·30	14·26	14·63	15·60	15·98	16·95	17·16	18·26	18·35	19·63	19·56	20·98
160	11·59	12·51	12·88	13·80	14·16	15·10	15·46	16·41	16·61	17·67	17·76	19·01	18·94	20·31
170	11·24	12·14	12·50	13·41	13·75	14·65	15·00	15·93	16·12	17·15	17·24	18·44	18·38	19·73
180	10·95	11·80	12·15	13·02	13·36	14·24	14·59	15·47	15·66	16·68	16·75	17·92	17·86	19·15
190	10·64	11·48	11·83	12·66	13·00	13·86	14·19	15·06	15·24	16·24	16·30	17·44	17·38	18·64
200	10·37	11·19	11·24	12·35	12·67	13·51	13·83	14·68	14·86	15 82	15·89	17·00	16·94	18·17
210	10·11	10·92	10·99	12·05	12·33	13·19	13·50	14·33	14·50	15·44	15·51	16·59	16·53	17·73
220	...	10·68	10·73	11·78	12·08	12·89	13·31	14·11	14·17	15·08	15·15	16·21	16·14	17·33
230	...	...	10·52	11·53	11·81	12·60	12·90	13·69	13·86	14·74	14·83	15·85	15·81	16·95
240	...	...	...	11 26	11·57	12·33	12·62	13·41	13·56	14·43	14·50	15·52	15·47	16·59
250	...	...	...	...	11·33	12·08	12·37	13·13	13·29	14·15	14·21	15·20	15·15	16·25
260	...	...	...	...	...	11·85	12·13	12·86	13·03	13·89	13·93	14·91	14·86	15·93
270	...	...	...	...	...	...	11·90	12·63	12·78	13 61	13·67	14 63	14·58	15 63
280	...	...	...	...	...	...	...	12·41	12·54	13·36	13·43	14·37	14·32	15·31
290	...	...	...	...	...	...	...	...	12·33	13·13	13·20	14·13	14·08	15·09
300	...	...	...	...	...	...	...	...	...	12·91	12·97	13·89	13·83	14·84

N.B.—For surface = pitch$^2 = \frac{a^2 + b^2}{2}$.

Table CIX.—Boiler Shells of 28 tons Tensile Steel and 80 per cent. Longitudinal Jointing: Working Pressure in lbs. per sq. inch allowed by B.M.E.D. & C. Committee.

Diameter of the Shell.	Thickness of Shell Plates in 32nds of an inch.													
	16.	17.	18.	19.	20.	21.	22.	23.	24.	25.	26.	27.	28.	29.
Feet.														
6·00	163	174	186	198	209	221	233	244	256	..	..	..	..	..
6·25	156	168	179	190	201	212	224	235	246	257	..	..	..	..
6·50	150	161	172	183	193	204	215	226	236	247	258	..	..	..
6·75	145	155	165	176	186	196	207	217	228	238	248	259	..	..
7·00	140	150	160	170	180	190	200	210	220	230	240	250	260	..
7·25	135	144	154	163	173	183	192	202	212	221	231	241	250	260
7·50	..	140	149	158	168	177	186	196	205	214	224	233	242	252
7·75	..	135	144	153	162	171	180	189	198	207	216	225	234	244
8·00	..	..	140	148	157	166	174	183	192	201	209	218	227	236
8·25	..	..	135	144	152	161	169	178	186	195	203	212	220	229
8·50	..	..	..	139	148	156	164	172	181	189	197	205	214	222
8·75	..	..	..	136	144	152	160	168	176	184	192	200	208	216
9·00	..	..	..	..	139	147	155	163	171	178	186	194	202	209
9·25	..	..	..	..	136	143	151	158	166	173	181	189	196	204
9·50	..	..	..	..	..	140	147	154	162	169	176	184	191	198
9·75	..	..	..	..	..	136	143	150	157	165	172	179	186	193
10·00	..	..	..	..	..	..	140	147	154	161	168	175	182	189
10·25	..	..	..	..	..	..	136	143	150	156	163	170	177	184
10·50	..	..	..	..	..	..	..	139	146	153	160	166	173	180

N.B.—The pressure allowed by the Register Societies and Board of Trade is 97 per cent. of the above.

Table CIXa.—Boiler Shells of 32 tons Tensile Steel and 84 per cent. Longitudinal Joints: Working Pressure in lbs. per sq. inch allowed by the B.M.E.D. & C. Committee.

Diameter of the Shell.	Thickness of Shell Plates in 32nds of an Inch.													
	22.	24.	26.	28.	30.	32.	34.	36.	38.	40.	42.	44.	46.	48.
Feet.														
8·0	209	230	251	272	293	..	..	..	..	..	..	..	..	..
8·5	197	217	237	256	276	296	..	..	..	..	..	..	..	..
9·0	186	205	223	242	260	279	298	..	..	..	..	..	..	..
9·5	176	194	212	229	247	265	282	300	..	..	..	..	..	..
10·0	168	184	201	218	235	252	268	285	302	..	..	..	..	..
10·5	160	176	192	208	224	240	256	272	288	304	..	..	..	..
11·0	152	167	183	198	213	229	244	259	274	290	..	..	..	..
11·5	146	160	175	189	204	219	233	248	262	277	292	..	..	..
12·0	..	154	168	182	196	210	224	238	252	266	280	294	..	..
12·5	..	147	161	174	187	201	214	228	241	255	268	281	295	..
13·0	..	..	155	167	180	193	206	219	232	245	258	271	283	296
13·5	..	..	149	161	174	186	199	211	223	236	248	261	273	286
14·0	..	..	144	156	168	180	192	204	216	228	240	252	264	276
14·5	..	..	..	151	162	174	185	197	209	220	232	243	255	267
15·0	..	..	..	145	156	168	179	190	201	213	224	235	246	257
15·5	..	..	..	..	151	162	173	184	195	206	216	227	238	249
16·0	.	..	..	..	147	157	168	178	189	199	210	220	231	241
16·5	..	..	..	..	142	152	163	173	183	193	203	214	224	234
17·0	..	..	..	..	..	148	158	168	178	188	197	207	217	227
17·5	..	..	..	..	..	144	153	163	173	182	192	202	211	221
18·0	..	..	..	..	..	..	149	158	168	177	186	196	205	214
18·5	..	..	..	..	..	..	145	154	163	172	181	191	200	209

N.B.—The pressure allowed by the Register Societies and Board of Trade is 97 per cent. of the above.

140. Hemispherical ends subject to internal pressure may be allowed double the pressure that is suitable for a cylinder of the same diameter and thickness.

141. Ends of steam receivers and similar vessels which are dished to a partial spherical form and flanged in a hydraulic press, the whole end being operated upon at each heat, may be passed without stays when complying with the following:—

Rule 253.

$$\text{Working pressure} = \frac{15 \times S(t-1)}{R};$$

t is the thickness of plate in 32nds of an inch.

R is the inner radius of curvature of the end in inches which shall not exceed the diameter of shell.

S is the minimum tensile strength of plate in tons.

When the end has a manhole in it, the plate must be thicker by $\frac{1}{32}$nds of an inch than given by rule.

The inside radius of curvature at the flange must be not less than 4 times the thickness of plate, and in no case less than 2·5 inches.

The total depth of flange when w is the minor axis in inches of the manhole $= 0{\cdot}177\sqrt{t \times w}$.

The above instructions do not apply to dished ends of vertical donkey boilers, which are subject to the thrust of the uptake in addition to the pressure of steam.

142. Stays, properly distributed, should be fitted to dished ends which are not of the thickness required for flat ends, or which do not comply with the requirements stated in Section 141; but, if the ends are sufficient for the pressure needed, when considered as portions of spheres, the stays, if of solid steel, may have a nominal stress of 18,000 lbs. per square inch of net section. If dished ends are not equal to the pressure needed when considered as portions of spheres, they should be stayed as flat surfaces.

LLOYD'S OLD RULES FOR FLAT SURFACES.

Table CX.—Lloyd's Constants for Flat Surfaces.

Description of attachment of stay.	Value of C.
Screw stays with riveted head,—	
Plates $\frac{7}{16}$-inch thick and under, . . .	90
Plates over $\frac{7}{16}$-inch thick, ...	100
Screw stays fitted with nuts,—	
Plates $\frac{7}{16}$-inch thick and under, . . .	110
Plates over $\frac{7}{16}$-inch and under $\frac{9}{16}$-inch, .	120
Plates $\frac{9}{16}$-inch and over, ...	135
Stays fitted with double nuts, ...	175
Stays fitted with double nuts and with washers of diameter equal to $\frac{1}{3}$rd the pitch, and of thickness equal to half that of the plate they cover, . .	185
Stays fitted with double nuts and with outside riveted washers of diameter equal to $\frac{2}{5}$ths pitch, and of thickness equal to half that of the plate they cover,	200
Stays fitted with double nuts and with outside riveted washers of diameter equal to $\frac{2}{3}$rds pitch, and of thickness equal to that of the plate they cover, .	220
Stays with double nuts and outside doubling strips of width equal to $\frac{2}{3}$rds distance between rows of stays, and thickness equal to that of the plate they cover, when P is taken as the pitch of rows, .	220
Stays with double nuts and outside doubling strips of width equal to $\frac{5}{8}$rds distance between rows of stays, and thickness equal to that of the plate they cover, when P is taken as the pitch of stays in the rows, ...	240

N.B.—Lloyd's Register has adopted the rules of the B.M.E.D. & C. Committee.

Table CXI.—Corrugated Furnaces (Fox, Morison, Deighton, Purves): Working Pressure in lbs. per sq. inch allowed by B.M.E.D. & C. Committee.

Diameter at Bottom of Corrugations.	Thicknesses in 32nds of an inch.													
	12.	13.	14.	15.	16.	17.	18.	19.	20.	21.	22.	23.	24.	25.
ins.														
30	176	192	208	224	240	256	272	288	304	...	...	...	...	...
31	170	185	201	216	232	247	263	278	294	309	...	...	...	...
32	165	180	195	210	225	240	255	270	285	300	...	...	...	...
33	160	174	189	203	218	232	247	261	276	291	305	...	...	...
34	155	169	183	197	211	226	240	254	268	282	296	310	...	...
35	150	164	178	192	205	219	233	246	260	274	288	301	...	...
36	146	160	173	186	200	213	226	240	253	266	280	293	306	...
37	143	156	169	182	195	208	221	234	247	260	273	286	299	...
38	...	151	164	176	189	202	214	227	240	252	265	277	290	303
39	...	147	160	172	184	196	209	221	233	246	258	270	283	295
40	...	144	156	168	180	192	204	216	228	240	252	264	276	288
41	...	...	152	163	175	187	199	210	222	234	245	257	269	281
42	...	...	148	160	171	183	194	205	217	228	240	251	263	274
43	...	...	145	156	167	178	189	200	212	223	234	245	256	267
44	...	...	...	152	163	174	185	196	207	218	229	240	251	261
45	...	...	...	149	160	170	181	192	202	213	224	234	245	256
46	...	...	...	146	156	166	177	187	198	208	219	229	240	250
47	...	...	...	...	153	163	174	183	194	204	214	224	235	245
48	...	...	...	...	150	160	170	180	190	200	210	220	230	240
49	...	...	...	...	147	156	166	176	186	196	206	215	224	234
50	...	...	...	...	144	153	163	172	182	192	201	210	219	229
51	...	...	...	...	...	150	160	169	178	188	197	207	216	225
52	...	...	...	...	...	147	157	166	175	184	193	203	212	221

N.B.—For the working pressure on a Leeds Forge bulb suspension furnace multiply by 1·065.

Lloyd's Rules for the wide spaces between nests of tubes.—p is the horizontal distance from centre to centre of the bounding rows of tube, and C as follows:—

Rule 255.

$$\text{Working pressure} = \frac{C \times t^2}{p^2};$$

t is in 16ths of an inch. Values of C are as follows:—

Pitch of Stay Tubes in bounding rows.	When tubes have no nuts outside plates.	When tubes are fitted with nuts outside plates.
Where there are two plain tubes between each stay tube, . . .	120	130
Where there is one plain tube between each stay tube, . . .	140	150
Where every tube in these rows is a stay tube, and nuts when fitted are on alternate tubes, . . .	160	170

Board of Trade, Lloyd's, &c., Rules for Compression on Tube Plates, &c.

The compressive load on tube plates shall be calculated by the following formula, in which the stress is taken at 14,000 lbs. per square inch.

Rule 255a.

$$\text{Working pressure} = \frac{875 \times (D - d) \times t}{W \times D},$$

where t is the thickness of plate in 32nds of an inch.

D is the horizontal distance apart of the tubes centre to centre, in inches.

d is the internal diameter of the plain tubes.

W is the width of combustion chamber measured inside from the tube plate to back chamber plate, or between the tube plates in double-ended boilers, with chambers common to opposite furnaces.

Girders supporting Combustion Chamber, Tops, &c.

Rule 258.

$$\text{Working pressure} = \frac{C \times d^2 \times T}{(L - P)D \times L} \times \frac{S}{28}.$$

B.M.E.D. & C. Committee rule for girders supporting combustion chamber, tops, &c.

Rule 258.

$$\text{Working pressure} = \frac{C \times d^2 \times T}{(L-P)\,D \times L} \times \frac{S}{28},$$

where L = width between tube-plates or tube-plate and back plate of chamber;
P = pitch of stays in girder;
D = distance from centre to centre of girder;
d = depth of girder at centre;
} All in inches.

T = thickness of girder at centre in *thirty-seconds of an inch.*

S is the minimum tensile strength in tons of the plates forming the girder. When it is a forging, S is 24 for iron and 28 for steel.

$C = \frac{n}{n+1} \times 495$ when the number of stays to each girder is odd;

$C = \frac{n+1}{n+2} \times 495$ when it is even; n being the number.

Board of Trade, &c., Rules relating to Stays.

Rule 259. Solid steel stays may be allowed by the Board of Trade a working stress with 28/32 ton steel up to 11,000 lbs. per square inch of net section, and on 26/30 ton steel to 9000 lbs.

Welded steel stays may not be used, but this does not apply to stay tubes.

When the threads of longitudinal stays are finer than six per inch, the depth of the external nuts should be at least 1¼ times the diameter of the stay.

144. Stay tubes made of steel which has been tested may be allowed a stress not exceeding 7500 lbs. per square inch of net section, provided that their net thickness is in no case less than ¼ inch.

Iron screwed stays in the combustion chambers resist the cross-breaking stresses, to which they are subjected, better than steel; a stress of 9000 lbs. per square inch may be allowed on such stays, provided the bars have been tested and have a tensile strength of not less than 21½ tons and an elongation, in 8 inches, of not less than 27 per cent.

155. A stress of 6000 lbs. may be allowed on the net section of iron stay tubes, provided that the net thickness is in no case less than ¼ inch.

156. The thickness of ordinary smoke tubes should not be less than:—

$$T = \frac{W.P. \times D}{9000} + \cdot 085,$$

where T = thickness of tube, in inches;
D = outside diameter of tube, in inches.

Screwed stays for combustion chambers, etc., may be of tested iron equal to 21·5 tons ultimate tensile strength, with an elongation of 25 per cent. on Standard test-piece B, and 30 per cent. on test-piece F; they may be of 26/30 tons tensile steel whose elongation is not less than 23 per cent. on Standard test-piece B, made by the Open Hearth Process, acid or basic. They shall be screwed with 9 threads to the inch, and accord with the following:—

Rule 259a.

$$\text{Working pressure} = \frac{(d - 0{\cdot}267)^2 \times 8250}{a},$$

where d is the diameter over the threads.

a is the area in square inches of plate supported by the stay.

The longitudinal and other stays not exposed to flame may be of the 28/32 Standard tensile steel; the number of threads per square inch shall be 6.

Rule 259b.

$$\text{Working pressure} = \frac{(d - 0{\cdot}340)^2 \times 9500 \times S}{28a}.$$

S is the minimum strength of the steel bar in tons per square inch. The stress on these stays must not exceed 11,000 lbs.

When the longitudinal stays are made with enlarged ends and the body of the stay is smaller in diameter than at the bottom of the thread, and where fewer threads than 6,

$$\text{the working pressure} = \frac{(d - 0{\cdot}125)^2 \times 9500 \times S}{28a}.$$

d is the diameter of thread at bottom or the smallest part of the stay body.

Table CXII.—Surface of Plate supported by one Screwed Stay, 9 Threads per inch.

Diameter over Threads.	Working pressures in pounds per square inch.											
	150.	160.	170.	180.	190.	200.	205.	210.	215.	220.	225.	230.	235.
1¼	54	51·0	48·0	45·4	42·4	40·8	39·8	38·9	38·0	37·1	36·4	35·6	34·7
1⅜	67·7	63·0	59·7	56·4	53·4	50·8	49·6	48·4	47·3	46·2	45·2	44·3	43·5
1½	83·6	78·3	73·7	69·6	66·0	63·1	61·2	59·7	58·3	57·0	55·8	54·7	53·6
1⅝	101	94·7	89·2	84·2	79·8	76·3	73·9	72·3	70·6	69·0	67·5	65·8	65·0
1¾	121	113	106	100	95·4	91·3	88·4	86·4	84·4	82·5	80·7	78·7	77·6
1⅞	142	133	125	118	112	107	104	101	99·1	96·7	94·7	92·4	91·1
2·0	165	155	146	138	130	125	121	118	115	112	110	108	106
2⅛	190	178	167	158	150	143	139	136	133	129	126	124	122
2¼	215	202	190	180	171	163	158	154	151	148	145	141	138
2⅜	244	229	215	203	193	184	178	174	170	167	163	159	156

Table CXIII.—Surface of Plate supported by one Stay of 28 tons Tensile Steel, 6 Threads per inch.

Diameter over Threads.	Working pressures in pounds per square inch.											
	150.	160.	170.	180.	190.	200.	205.	210.	215.	220.	225.	230.	235.
2	175	164	154	145	138	131	128	124	121	119	116	114	111
2 1/8	201	189	178	168	159	151	147	144	141	137	134	131	127
2 1/4	231	216	204	192	182	173	168	164	161	158	154	151	147
2 3/8	262	246	232	217	207	196	191	187	183	179	175	171	167
2 1/2	295	277	261	246	233	221	216	211	206	201	197	193	189
2 5/8	330	310	292	275	261	248	241	236	231	225	220	216	212
2 3/4	368	345	325	306	290	276	269	262	256	251	245	240	236
2 7/8	406	382	360	339	321	305	297	290	284	278	271	266	261
3	448	420	396	373	354	336	328	320	313	306	299	293	287
3 1/8	491	460	434	409	388	368	359	350	342	335	327	321	315
3 1/4	..	..	473	447	423	402	392	383	374	367	358	350	343
3 3/8	..	..	..	..	460	437	426	416	407	398	389	380	372
3 1/2	..	..	..	..	..	474	462	451	441	431	421	412	403

For surfaces that may be supported by stays other than screw stays and exceeding 1½ inches smallest diameter, see Table CXIII.

The number of threads per inch for boiler stays has been established by the Engineering Standards Committee to be 9 per inch for stays above 1¼ inches diameter, and 6 per inch for those above 2 inches, fitted with nuts on both sides of the plate.

RULES FOR FURNACES.

The following rules are those followed by the Board of Trade, Lloyd's, and the other Register Societies.

The working pressure to be allowed on *plain furnaces* strengthened by the Adamson or other joints, and on the cylindrical *bottoms of combustion chambers* is to be obtained as follows, the least pressure by either formula being taken :—

Rule 261a.

$$\text{Working pressure} = \frac{C(t-1)^2}{(L+24)\times D}.$$

Rule 262b.

$$\text{Working pressure} = \frac{C_1}{D} \times 10(t-1) - L.$$

D is the external diameter in inches.

t is the thickness of furnace in 32nds of an inch.

L is the length between points of substantial support in inches measured from centres of rivet rows, or from the commencement of flange curvature.

C is 1450 when the longitudinal seams are welded, 1300 when they are riveted.
C_1 is 50 when welded, and 45 when riveted.

Corrugated Furnaces.

Rule 263. Working pressure $= \frac{C(t-1)}{D}$.

D is the external diameter measured at the bottom of the corrugation in inches.
t the thickness in 32nds of an inch at that point.
C is 480 for the Fox, Morison, Deighton, Purves, and other similar furnaces. For the Leed's Forge Bulb Suspension furnace, C is 510.

In no case should the thickness of any furnace exceed $^{26}/_{32}$nds of an inch.

Standard Furnaces.

By the British Engineering Standards the following is the practice with makers of furnaces :—

All furnaces shall be made of steel produced by the Open Hearth Process, acid or basic, having a maximum tensile strength of 30 tons, and a minimum of 26 tons per square inch, with an elongation not less than 23 per cent. in 8 inches.

The criterion diameter is that inside the furnace at bottom of corrugation (that is the minimum inside diameter).

The diameter at the mouth outside shall be $5\frac{1}{2}$ inches larger for the Morison and Deighton, and $6\frac{1}{4}$ inches for the Leeds Forge Suspension. The pitch of these corrugations is 8 inches ; the plain part at the front end may be 10·5, 8·5, 6·5, or 4·5 inches. At the back end from the last ridge to the front of flange 10·5 inches.

The standard lengths of these furnaces over all will be then a multiple of 8 inches plus 21, 19, 17, or 15 inches.

The Standard furnaces are made from 36 inches minimum internal diameter to 48 inches, varying by increments of 1 inch. The Bulb Suspension vary from 35 inches to 47 inches.

Smoke tubes standard lengths over the tube-plates are 3 inches in excess of the length of the furnaces ; the tubes are 1 inch longer, that is the plain tubes are 4 inches longer than the furnaces over all.

Stay tubes can now be obtained with swelled ends at a less price than what rules for the plain tube with a minus thread at the back end ; moreover, they are much lighter. Their diameter in the body is the same as that over the thread at back end. The thread at front end is $\frac{1}{4}$ inch greater in diameter than that at the back. If d is the standard diameter of the plain smoke tubes and also that over the threads at back end of the stay tube, then

Diameter over thread at front end $= d + \frac{1}{4}$ inch.

The Board of Trade and the Register Societies require that the minimum thickness of stay tubes under the threads shall be $\frac{1}{4}$ inch for marginal, and $\frac{3}{16}$ inch for all others.

The threads shall be continuous and 9 to the inch.

The plain tubes shall be swelled at the front ends by $\frac{1}{8}$ inch in diameter for $2\frac{1}{2}$ inches in the length.

Stay tubes shall be screwed at the back end for 2 inches and at the front end 2 inches when without, and $2\frac{1}{2}$ inches with nuts.

Number of tubes possible in a given area and their surface:

(*a*)	48 inches square will permit		169	tubes	3·5	inches pitch.
(*b*)	49·25	,, ,,	144	,,	3·875	,,
(*c*)	48·375	,, ,,	121	,,	4·125	,,
(*d*)	47·0	,, ,,	100	,,	4·375	,,

From this the following holds good as the surface per foot:—

With tubes $2\frac{1}{2}$ inches external diameter, a square foot of plate will accommodate tubes giving 6·91 square feet of surface per foot of length. Tubes $2\frac{3}{4}$ inches diameter will give 6·15 square feet. Tubes 3 inches diameter 5·84 square feet, and those $3\frac{1}{4}$ inches 5·54 square feet. If, however, the $2\frac{1}{2}$-inch tubes have a pitch of $3\frac{5}{8}$ inches, the surface per foot will be 6·46 square feet.

RULES RELATING TO TESTING BOILERS GENERALLY.

Hydraulic tests.—New boilers should be tested by hydraulic pressure before being lagged or placed in the vessel. The test pressure to remain on for at least ten consecutive minutes in all cases.

Board of Trade, Lloyd's, and other Registries.—Twice the working pressure for boilers whose working pressure does not exceed 100 lbs. Over 100 lbs. W.P. the test is $1{\cdot}5 \times \text{W.P.} + 50$ lbs.

Boilers which have been in use are not to exceed $1{\cdot}5 \times \text{W.P.}$

British Admiralty.—W.P. + 90 lbs.

German Authorities.—W.P. + 75 lbs.

French Authorities.—W.P. + 85 lbs.

EVAPORATORS, &c. (Board of Trade Rules).

164. Evaporators, generators, feed make-ups, &c., where the evaporation of water under pressure is an essential feature, should be regarded as steam boilers, whether the evaporation is effected by heat from coal gas, from steam, or from any other source, and they should be examined by the Surveyor on each occasion the vessel is surveyed for passenger certificate in the same manner as other boilers on board the vessel; and the particulars regarding them, their safety-valves, &c., should be recorded on the declaration in the same manner as is done in the case of other auxiliary boilers.

The strength, quality of material, and method of construction of such apparatus, should, as a rule, be in accordance with the regulations for steam boilers. For moderate pressures, however, evaporators of the

type usually fitted may be made of cast material, but in no case should the pressure exceed 15 lbs. per square inch when the main body of the apparatus is a single casting. Subject to this limitation and to the sanction of the Board for the use of such material in each particular case, evaporators made of cast-iron, or of gun-metal, having a tensile strength of not less than 10 tons per square inch, may be allowed a working pressure not exceeding that found by the following formulæ, provided the thickness is not less than $\frac{5}{8}$ inch in the case of cast-iron, and $\frac{3}{4}$ inch in the case of gun-metal, and the castings are in every way sound and to the Surveyor's satisfaction :—

Cylindrical Shells.

$$\frac{C(T-\frac{1}{8})}{D} = \text{working pressure.}$$

Circular Flat Surfaces.

$$\frac{C_1 \times T^2}{D^2} = \text{working pressure.}$$

Square Flat Surfaces.

$$\frac{C_2 \times T^2}{S^2} = \text{working pressure.}$$

Where T = thickness, in inches ;
D = diameter, in inches ;
S = side, in inches ;
C for cast-iron = 4,000 ;
C_1 ,, ,, ,, = 24,000 ;
C_2 ,, ,, ,, = 16,000 ;
C ,, gun-metal = 6,000 ;
C_1 ,, ,, ,, = 30,000 ; and
C_2 ,, ,, ,, = 20,000.

If cast-steel is used, the minimum thickness should not be less than $\frac{1}{2}$ inch, and the constants C, C_1, and C_2 may be 10,400, 52,000, and 34,700 respectively.

The formulæ may also be used for determining the working pressure permissible for feed heaters, feed filters, &c., and for such vessels the constants for cast-iron and cast-steel may be increased by 25 per cent.

Where there are large branches, doors, or other large openings in the castings, the scantlings will require to be materially increased, and such cases should be submitted for consideration before being passed.

When the ends are cast solid with the shell, there should be a substantial fillet all round inside, and when the ends are bolted, D, in the flat surface formula, should be the diameter of the bolt circle. The flange should be of sufficient thickness and have a substantial fillet at the root all round.

165. In calculating the strength of studs or bolts securing the covers of evaporators, feed heaters, feed filters, and other similar vessels, the

calculated load on the cover should be found by multiplying the area of the pitch circle by the working pressure, and the nominal stress allowed on the net section of the material of the studs, &c., should not exceed 7000 lbs. per square inch in the case of iron, and 9000 lbs. per square inch in the case of steel, the maximum allowance being permissible only when the diameter is $\frac{7}{8}$ inch or more. When the studs or bolts are of a diameter less than $\frac{7}{8}$ inch, the nominal stress allowed per square inch of net section should not exceed 6000 lbs. in the case of iron, and 7200 lbs. in the case of steel, owing to the relatively greater stress to which they are subject when the nuts are tightened.

Studs or bolts securing covers which are required to be frequently removed should not be less than $\frac{1}{8}$ inch in diameter.

166. The mountings, &c., for evaporators should, as a general rule, be similar to those required in the case of boilers on board passenger vessels, but a single safety-valve may be allowed, provided it is of sufficient size.

167. When a reducing nozzle is fitted in the steam supply pipe, the contracted orifice should not, in ordinary cases, exceed that found by the following formula:—

$$\frac{A \times p}{6 \times P} = \text{area of orifice},$$

where A = combined area of safety-valves fitted to the evaporator;
p = absolute pressure at which the evaporator is worked; and
P = absolute pressure of entering steam.

The reduced orifice for an evaporator having only one safety-valve should not, in any case, be greater than would be allowed, for the same pressures, with a single valve $2\frac{1}{16}$ inches in diameter (*i.e.* equivalent in area to two 2-inch valves).

Reducing orifices should be bored through brass or similar material, and should be parallel for a length of at least $\frac{1}{4}$ inch; and each nozzle should be formed with a facing at the side, on which particulars regarding the safety-valves, their load, the maximum pressure of the entering steam, and the diameter of the orifice should be stamped, as shown by the following example:—

2 S.V. DIA. 3". LOAD 10 lbs. sq. in.
BOILER PRESS. 160 lbs.
REDUCED ORIFICE 21/32" DIA.

168. On the completion of the hydraulic test of an evaporator, a feed heater, feed filter, or other similar vessel which complies with the Board's regulations and has been surveyed during construction, the Surveyor, in order to afford means of identification, should stamp the

apparatus in a conspicuous place, with the pressure applied to the shell, the pressure applied to the coils (if any), the date, and his initials.

169. Feed pipes, feed heaters, filters or other vessels through which the feed water passes on its way from the pumps to the boilers, should be made sufficient for a pressure 20 per cent. in excess of the boiler pressure; and an efficient relief-valve (or valves), suitably adjusted, and of a type which does not present a ready means of overloading, should be fitted where required to prevent this pressure being exceeded in any part of the feed system, under any conditions likely to occur. In cases where the main feed pumps are independent of the main engines, the area of the relief-valve (or valves) should not, as a rule, be less than half that of the feed discharge pipe, provided the latter is sufficient to ensure a reasonable velocity of flow in the pipe; but, when the main pumps are worked by the main engines, the relief-valve (or valves) should be of at least the same area as the pipe.

Local heating of plates should be avoided.

Minimum thickness of plates to be used, $\frac{1}{16}$ inch.

Annealing.—All plates which have been punched, flanged, or locally heated, and all stays and stay tubes which have been locally heated, must be carefully annealed afterwards.

Welding.—Steel plates that have been welded should not be passed to carry a tensile stress; when subject to a compressive stress they should be efficiently annealed after welding.

Boiler tracings, &c.—Tracings of boilers may be received for examination upon payment of the usual fee of £2, and the Surveyors may proceed as far as witnessing the hydraulic test before any further instalment of the survey fee is paid. Engineers and boilermakers should be advised of this arrangement.

Donkey boilers that are in any way attached to, or connected with, the main boilers, or with the machinery used for propelling the vessel, should be surveyed and have their working pressure fixed in the same way as the main boilers, and have a water and steam gauge, and all other fittings complete, and, as regards safety valves, should comply with the same regulations as the main boilers.

Launch boilers.—The boilers of steam launches forming part of the statutory boat capacity of passenger steamers should as regards construction, strength, material, safety valves, and other fittings comply with the same regulations as the main boilers.

LLOYD'S RULES RELATING TO STEEL BOILERS GENERALLY.

Boilers.

Section 3. 1. The Surveyors will be guided in fixing the working pressure by the tables and formulæ annexed. (*See* section 11.)

2. Any novelty in the construction of the machinery or boilers to be reported to the Committee.

3. The boilers, together with the machinery, to be inspected at different stages of construction.

All the holes in steel boilers should be drilled, but if they be punched the plates are to be afterwards annealed.

All plates that are dished or flanged, or in any way heated in the fire for working, except those that are subjected to a compressive stress only, are to be annealed after the operations are completed.

No steel stays are to be welded.

Unless otherwise specified, the Rules for the construction of iron boilers will apply equally to boilers made of steel.

4. The boilers to be tested by hydraulic pressure, in the presence of the Engineer-Surveyor, to twice the working pressure, and carefully gauged while under test.

5. Two safety valves to be fitted to each boiler, and loaded to the working pressure in the presence of the Surveyor. In the case of boilers of greater working pressure than 60 lbs. per square inch, the safety valves may be loaded to 5 lbs. above the working pressure. If common valves are used, their combined areas to be at least half a square inch to each square foot of grate surface. If improved valves are used they are to be tested under steam in the presence of the Surveyor; the accumulation in no case to exceed 10 per cent. of the working pressure.

6. An approved safety valve also to be fitted to the superheater.

7. In winch boilers one safety valve will be allowed, provided its area be not less than half a square inch per square foot of grate surface.

8. Each valve to be arranged so that no extra load can be added when steam is up, and to be fitted with easing gear which must lift the valve itself. All safety-valve spindles to extend through the covers and be fitted with sockets and cross handles, allowing them to be lifted and turned round in their seats, and their efficiency tested at any time.

9. Stop-valves to be fitted so that each boiler can be worked separately.

10. Each boiler to be fitted with a separate steam gauge, to accurately indicate the pressure.

11. Each boiler to be fitted with a blow-off cock independent of that on the vessel's outside plating.

12. The machinery and boilers are to be securely fixed to the vessel to the satisfaction of the Surveyor.

RIVETED JOINTS.

Board of Trade.—The following sketches of riveted joints, and formulæ for determining their various proportions, are given in an appendix to the Board of Trade Rules; the formulæ are here given in a form differing slightly from that adopted in the Rules:—

F in the following formulæ stands for the factor of safety (for which see Table CIV.), r for percentage of plate left between rivet holes, and S_1 for tensile strength of plates in tons.

ORDINARY CHAIN AND ZIGZAG RIVETED JOINTS.

$$\left.\begin{matrix}\text{Percentage of plate}\\ \text{left between holes}\end{matrix}\right\} = \frac{100\ (\text{pitch} - \text{diameter of rivet})}{\text{pitch}} = r.$$

$$\left.\begin{matrix}\text{Nominal per-}\\ \text{centage of}\\ \text{rivet section}\end{matrix}\right\} = \frac{\left.\begin{matrix}958{\cdot}33\ \text{Butt}\\ 511{\cdot}11\ \text{Lap}\end{matrix}\right\} \times \text{area of riv.} \times \text{No. rivets in pitch} \times F}{S_1 \times \text{pitch} \times \text{thickness of plate}}.$$

Rule 267. To find pitch so that nominal rivet section and net plate section may be of equal strength:—

$$\text{Pitch} = \frac{\left.\begin{matrix}958{\cdot}33\ \text{Butt}\\ 5{\cdot}111\ \text{Lap}\end{matrix}\right\} \times \text{area rivet} \times \text{No. rivs. in pitch} \times F}{S_1 \times \text{thickness of plate}} + \text{dia. rivet.}$$

Rule 267a. To find pitch and diameter of rivet:—

$$\left.\begin{matrix}\text{Diameter}\\ \text{of rivet}\end{matrix}\right\} = \frac{S_1 \times r \times \text{thickness of plate}}{\left.\begin{matrix}7{\cdot}526\ \text{Butt}\\ 4{\cdot}014\ \text{Lap}\end{matrix}\right\} \times (100 - r) \times \text{No. rivets in pitch} \times F}.$$

$$\text{Pitch} = \frac{\left.\begin{matrix}13{\cdot}29\ \text{Butt}\\ 24{\cdot}911\ \text{Lap}\end{matrix}\right\} \times S_1 \times r \times \text{thickness of plate}}{(100 - r)^2 \times \text{No. rivets in pitch} \times F}.$$

Also, when diameter of rivet is found first:—

Rule 268. $$\text{Pitch} = \frac{100 \times \text{diameter of rivet}}{100 - r}.$$

When double butt straps are used each strap must have a thickness of $\frac{5}{8}$ths of the plate it covers.

A single butt strap must have a thickness equal to $1\frac{1}{8} \times$ thickness of plate it covers.

Distance from centre of rivet to edge of plate (joint E) $= 1\frac{1}{2} \times$ diameter of rivet.

Distance between rows of rivets:—

(*a*) **Chain riveted joints** (figs. B, C, E_2, G),—

Rule 269.

$V =$ not less than $2 \times$ dia. of rivet, preferably $\frac{(4 \text{ dia. rivet}) + 1}{2}$.

(*b*) **Zigzag riveted joints** (figs. B_1, C_1, E_1, G_1),—

$$V = \frac{\sqrt{(11 \text{ pitch} + 4 \text{ dia. rivet})\ (\text{pitch} + 4 \text{ dia. rivet})}}{10}.$$

Diagonal pitch (figs. B_1, C_1, E_1, G_1),—

$$p_D = \frac{6 \text{ pitch} + 4 \text{ dia. rivet}}{10}.$$

RIVETED JOINTS WITH ALTERNATE RIVETS IN OUTER, OR OUTER AND INNER, ROWS LEFT OUT.

$$\text{Percentage of plate left between holes} = \frac{100 \text{ (pitch - dia. of rivet)}}{\text{pitch}}$$

$$\left.\begin{matrix}\text{Nominal per-}\\ \text{centage of}\\ \text{rivet section}\end{matrix}\right\} = \frac{\left.\begin{matrix}958{\cdot}33 \text{ butt}\\ 511{\cdot}11 \text{ lap}\end{matrix}\right\} \times \text{area of riv.} \times \text{No. rivets in pitch} \times F}{S_1 \times \text{pitch} \times \text{thickness of plate}}.$$

Double butt straps for this type of joint must each be of thickness given by,—

Rule 270.

$$\text{Thickness of butt strap} = \frac{5 \times \text{thick. of plate} \times (\text{pitch} - \text{dia. of rivet})}{8 \times (\text{pitch} - 2 \text{ dia. of rivet})}.$$

Distance from centre of rivet to plate edge (joint E, p. 383) = $1\frac{1}{2}$ × diameter of rivet.

Rule 271. Distance between rows of rivets :—

(*a*) **Chain riveted joints** (H, p. 383),—

$$\left.\begin{matrix}V = \sqrt{\frac{(11 \text{ pitch} + 4 \text{ dia. rivet})(\text{pitch} + 4 \text{ dia. rivet})}{10}}\\ \text{or } V = \text{not less than } 2 \times \text{dia. of rivet, and preferably } \frac{(4 \times \text{dia. rivet}) + 1}{2}\end{matrix}\right\} \begin{matrix}\text{The greater}\\ \text{of these}\\ \text{two values}\\ \text{to be used.}\end{matrix}$$

(*b*) **For joint H** (p. 383),—

$V_1 = 2 \times$ dia. of rivet as a minimum, but the value $\frac{(4 \times \text{dia. of rivet}) + 1}{2}$ is preferable.

(*c*) **Zigzag riveted joints** (H_1, p. 383),—

$$V = \sqrt{(\tfrac{11}{10} \text{ pitch} + \text{dia. rivet})(\tfrac{1}{10} \text{ pitch} + \text{dia. rivet})}.$$

Diagonal pitches :—

(*a*) **Diagonal pitch** (H_1, p. 383),

$$p_D = \tfrac{8}{10} \text{ pitch} + \text{dia. of rivet.}$$

(*b*) For joint H_1 (p. 383),—

$$P_D = \frac{3 \text{ pitch} + 4 \text{ dia. rivet}}{10}.$$

Distance between inner rows of rivets (joint H_1, p. 383),—

$$V_1 = \frac{\sqrt{(11 \text{ pitch} + 8 \text{ dia. rivet})(\text{pitch} + 8 \text{ dia. rivet})}}{20}.$$

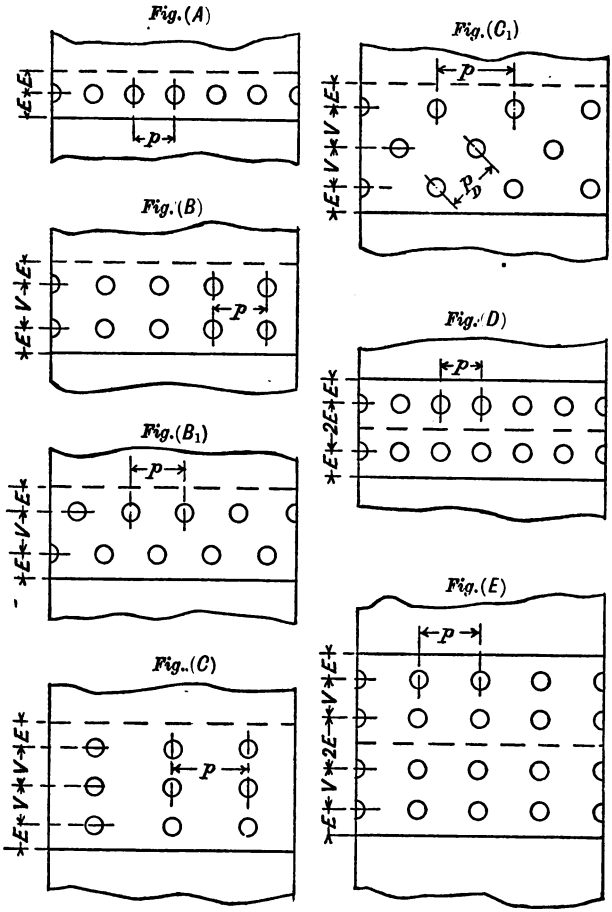
Maximum Pitches for Riveted Joints.

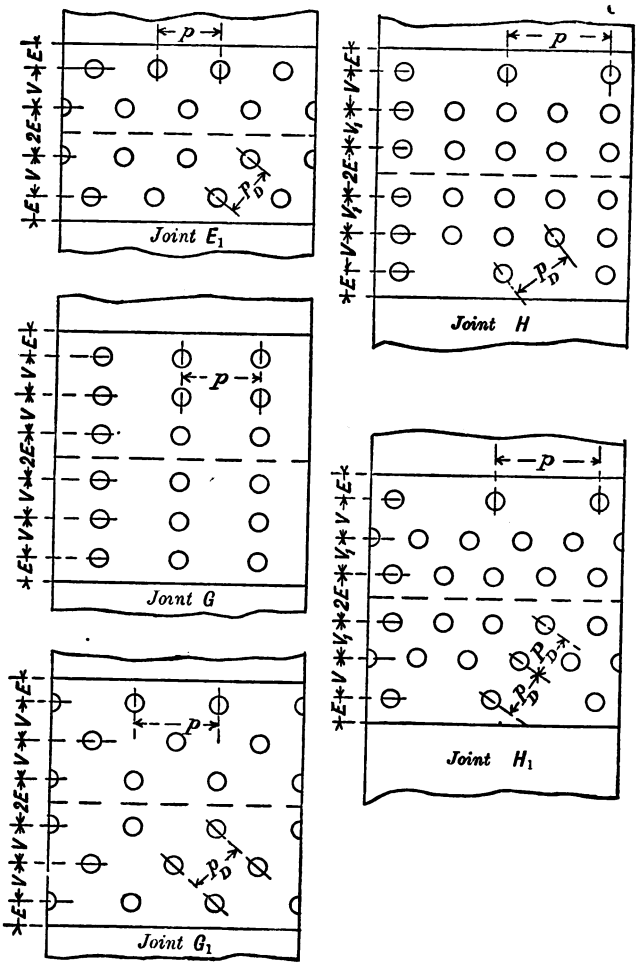
T = thickness of plate in inches;
p = maximum pitch of rivets in inches, provided it does not exceed 12½ inches; and
C = constant applicable from the following table :—

Number of Rivets in one Pitch.	Constants for Lap Joints.	Constants for Double Butt-Strap Joints.
1	1·31	1·75
2	2·62	3·50
3	3·47	4·63
4	4·14	5·52
5	...	6·00

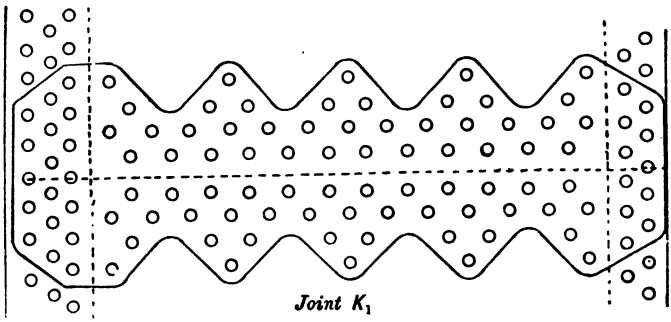
$$(C \times T) + 1 = p.$$

Diagrams of Board of Trade Rules.

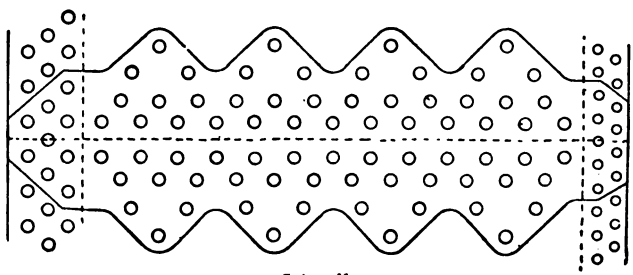



Joint E_1
Joint H
Joint G
Joint H_1
Joint G_1
p
p_d

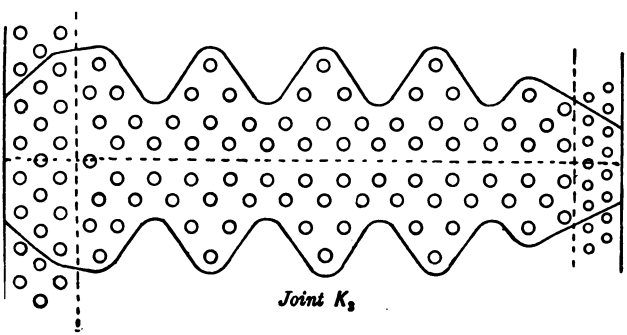
The following figs. are examples of quadruple riveting of the special kinds adopted largely on the Continent for boilers having thick shells.



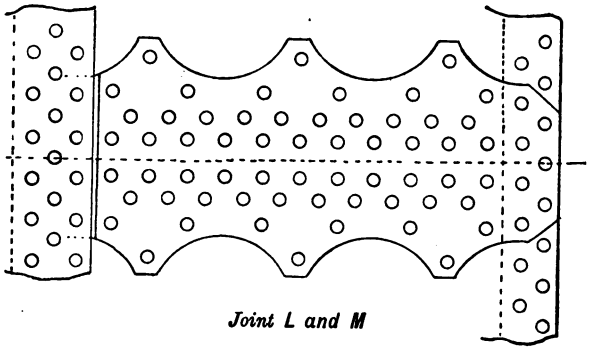
Joint K_1



Joint K_2

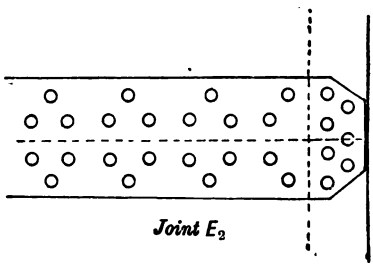


Joint K_3



Joint L and M

Joint E_2 is an example of a double riveted joint with alternate rivets outer rows omitted :—



Joint E_2

Table CXIV.—Tests of Boiler Material (Steel).

Part of Boiler.	Admiralty.		Board of Trade.		Lloyd's Register.		British Corporation.		Bureau Veritas.	
	Ultimate Strength.	Extension.	Ultimate Strength.	Extension.	Ultimate Strength.	Extension.	Ultimate Strength.	Extension.	Ultimate Strength.	Extension.
		Per cent.		Per cent.		Per cent.		Per cent.		Per cent.
Shell plates, Tons per sq. in.	27 to 30	20	28 to 32	20	28 to 32	20	28 to 32	20	to 30	20
Plates exposed to flame, "	24 to 27	25	26 to 30	20	26 to 30	20	26 to 30	20	to 28	22
Rivet bars, "	24 to 27	25	26 to 30	25	26 to 30	20	25 to 30	20	24 to 27	25
Stay bars, "	27 to 30	20	28 to 32	20	28 to 32	20	28 to 32	20	to 30	20
" " screwed, "	..	..	26 to 30	23	26 to 30	23	26 to 30	23	..	..
Steam-pipe plates, "	24 to 27	33	to 28	25	26 to 30	20	to 28	25	..	..
Corrugated furnaces, "	23 to 25	27	26 to 30	23	26 to 30	23	25 to 29	23	to 25	25
Tubes, . . .	26	27	to 28	25	to 28	25	to 28	25	..	..

Table CXV.—Various Kinds of Jointings for Plates and their Relative Values when designed in accordance with Board of Trade Rules.

Mark.	Description of Joint.	Equal to per cent. of Solid Plate.	Relative Working Pressure.
A	Lap joint, single riveted . . .	56	105
B	,, double ,, . . .	66	126
C	,, treble ,, . . .	72	140
D	Butt joint, with double butt straps, single riveted	66	126
E	Butt joint, with double butt straps, double riveted.	75	150
E_2	Butt joint, double straps, double riveted, half number in outer rows .	81/84	162/168
G	Butt joint, double straps, treble riveted	80/82	160/164
H	Butt joint, double straps, treble riveted, half number in outer rows .	84/88	168/176
J	Butt joint, double straps, quadruple riveted	84	168
K	Butt joint, double straps, quadruple riveted, special, 9 to the pitch . .	91·8	183·6
L	Butt joint, double straps, quadruple riveted, special, 11 to the pitch .	93·2	186·4
M	Butt joint, double straps, quadruple riveted, special, 11 to the pitch .	94·2	188·4

N.B.—In cases E_2, G, and H the higher percentage is obtained by using rivets large in diameter compared with the thickness of plates. Case E_2 is only suitable for plates comparatively thin, say up to $\frac{1}{4}$ inch, as the rivets have to be $1\frac{1}{2}$ times the thickness for steel of 27 tons tensile, and consequently, if 36-ton steel is used, the diameter will be excessive and the pitch too great for tight work.

In case M the very high percentage is obtained by using rivets of diameter 1·125 times the thickness of the 27-ton plate, as against the simple $d = t$ for 93·2 per cent.

SUPERVISION OF BOILER WORK.

Admiralty.—The following instructions to boilermaker overseers are those that were given in Admiralty specifications for cylindrical boilers:—

The boilers will be subject to the supervision of an overseer, who

will be directed to attend on the premises of the contractors during the progress of the work on the boilers, to examine the material and workmanship used in their construction, to witness the prescribed tests, and to see that this specification, as regards the boilers and work in connection, is conformed to in all respects by the contractors. The extent of supervision is described in the following paragraphs extracted from Admiralty instructions to overseers, and the contractors are to afford him every facility for their proper execution.

The plates and other material used in the construction of the boilers to be subjected to such tests as may be directed in the specification. Every plate used is to be carefully examined by the overseer for laminations, blisters, veins, and other defects, and to ensure that it is of the proper thickness and brand. No plate, angle, &c., which from any cause is considered by the overseer to be unfit for the intended use is to be fitted.

During the construction of the various parts of the boilers, the overseer is to satisfy himself that the dimensions as shown on the approved drawings are being adhered to by the contractors.

Whenever plates are flanged or welded, or in any case where iron or steel is worked in such manner that it is particularly liable to suffer in strength unless carefully handled, the overseer is to be present if possible on all occasions during the time the work on each article is in progress, and he is to fully satisfy himself that it is sound before he allows any part to be put in the boilers.

Samples of the rivets being used for the boilers are to be taken by the overseer during the progress of the work and tested as specified hereafter, and any batches of rivets found defective are to be rejected. Before rivets are put in, the overseer is to see that the plates are brought properly together, and that the holes are fair with one another. He is not to allow drifting on any account, but he is to see that they are carefully rimed fair where necessary. He is also to make sure during the progress of the work that the rivets fill the holes completely, and that the heads are properly set up, well formed, and finished.

The overseer is to see that all internal parts of the boiler are riveted with rivets having heads and points of approved shape, and that any seams he considers necessary are riveted on the fire side. No snap heads are to be allowed in the internal parts. Any proposal for hydraulic riveting the internal parts is to be submitted to the Admiralty, with sketch of the proposed heads and points. In all parts where the rivets are not closed by hydraulic riveting machinery, he is to see that the rivet holes are countersunk and that coned rivets are used. All holes in the plates, angles, &c., are to be drilled, and not punched, and are to be drilled in place after bending. The clearance between rivet hole and rivet before closing is not to be greater than approved by the overseer.

The overseer will see that the particulars of the form, dimensions, and pitch of the various stays shown on the drawings are adhered to, and samples of them are to be tested as directed in this specification; and he will be guided by his experience as a workman in testing and

judging of the soundness of the forging and construction of the various stays.

He is to see that palm stays if fitted are forged from the solid and not welded, that all short stays are nutted on all flat surfaces except where otherwise approved and screwed to a pitch of eight threads per inch for stays of 1 inch diameter and above, that the holes for the screwed stays in the water spaces are drilled and tapped together after the furnaces and combustion chambers have been riveted in place in the boiler, that the combustion-chamber stays are drilled square to the bevel of the combustion-chamber plates, and that no bevel washers are inside the chamber. Any girder stays used for combustion chambers are to be well bedded on to the tube plates to the satisfaction of the overseer.

The overseer is to see that the arrangement of the zinc plates shown on the approved drawings is adhered to, that the metallic surfaces in contact are filed bright, and that means are adopted to secure a firm grip of the clips by which the plates are attached.

The overseer is to witness the testing, in all cases, of the boiler tubes, in accordance with this specification, before they are put in the boiler.

When the boilers are reported to the overseer by the contractors as being completed, ready for testing by water pressure, the overseer is to witness a preliminary test of them in accordance with the specification, carefully observing with the assistance of gauges and straight edges whether any bulging or deflection of the plates has taken place.

The official test will be conducted on all occasions in the presence of an inspecting officer. A test pressure gauge is supplied to the overseer from the Admiralty, and the official test is to be made with this gauge.

After the boilers have been tested by water pressure the overseer is to see that they are properly cleaned inside and outside, and then well painted outside with red lead. It is important that the whole surface of the boilers should be thoroughly cleansed of scale formed in manufacture before any paint is put on them. The boilers are not to be exposed to the weather till they are so painted, and properly cleaned and closed up to his satisfaction.

The overseer is to make himself fully acquainted with the progress of the whole of the work in its various stages, to satisfy himself that every part is sound before it is allowed to be put in the boilers, and to see that the following instructions for the treatment of mild steel are strictly complied with.

Treatment of mild steel.—All plates or bars which can be bent *cold* are to be so treated; and if the whole length cannot be bent cold, heating is to be had recourse to over as little length as possible.

All plates of the boilers are to be flanged by hydraulic pressure, and in as few heats as possible.

In cases where plates or bars have to be heated, the greatest care should be taken to prevent any work being done upon the material after it has fallen to the dangerous limit of temperature known as a "blue heat," say from 600° to 400° F. Should this limit be reached during working, the plates or bars should be reheated.

Plates or bars which have been worked locally while hot are to be subsequently annealed over the whole of each plate or bar.

All plates for boilers and steam pipes and all tubes are to be treated as follows for removal of scale:—Previous to work being commenced on them, they are to stand for eight hours in a mixture of 19 of water to 1 of hydrochloric acid. They should be placed on edge and not laid flat. On removal from acid bath, they should be thoroughly brushed and washed in fresh water, and then placed on edge to dry.

N.B.—A shorter time in a somewhat stronger solution is safer, as it precludes the occluding of hydrogen and thereby making the metal brittle.

BOILER MOUNTINGS, &c.

Size of main stop-valves should be such that the whole of the steam is just capable of passing through without appreciable loss of pressure and no more. The lift of valve should be no more than will permit of this, for then, if with careless stoking or other causes the steam drop, it tends to cause priming. It is surprising how small a hole in a diaphragm, interposed between the steam-pipe flanges, will pass all the steam a boiler of large size can make. Such a diaphragm may be used often with advantage on boilers which are in the hands of careless or unskilled engineers, as it cannot be tampered with as can be the stop-valve.

The flow of steam through the lengths of pipe on ship-board may be taken as 6000 feet per minute in pipes under 2·5 inches diameter, 7000 up to 4 inches, 8000 up to 7 inches, 9000 to 12 inches and over 12 inches, 9500 to 15 inches per pressure about 700 lbs. per square inch. That is, velocity of boiler steam $= 5000 \times \sqrt[4]{\text{diameter}}$.

Rule 272. Area steam-pipe section $= \frac{\text{T.H. surf.}}{p} \times f + 3$ sq. in.,

for turbines, $f = 1{\cdot}25$; for quick-revolution reciprocators, 1·55; for ordinary mercantile marine reciprocators, 1·75.

Rule 273. Area of steam-pipe section $= \frac{\text{I.H.P.}}{p} \times \frac{c}{x} + 3$ sq. ins.,

c is the consumption of steam per I.H.P. hour, as given in Table XCIII.; x is $= 6{\cdot}2$ for turbines; $x = 5{\cdot}1$ for quick-revolution reciprocators, and $x = 4{\cdot}3$ for slow-running mercantile engines.

The area of stop-valve should be not less than 10 per cent. greater than the pipe to provide equal area clear of obstructions, that is,

Rule 274. Diameter of stop-valve $= 1{\cdot}06 \times$ diameter of steam-pipe.

If in certain cases the boilers are going to be forced so that the production is more than 8 lbs. per square foot of total heating surface, the factor f must be increased proportionately.

The following Tables will be found useful as a guide in determining the size of steam-piping from the demand requirements.

Table CXVI.—Quantity of Steam delivered per minute through Pipes whose length does not exceed 240 × √diameter under Service Conditions.

Diameter of Bore.	Live Steam, 200 lbs. Absolute.		Exhaust Steam, 3 lbs. Absolute.		Diameter of Bore.	Live Steam, 200 lbs. Absolute.		Exhaust Steam, 3 lbs. Absolute.	
	Cubic Feet.	Lbs.	Cubic Feet.	Lbs.		Cubic Feet.	Lbs.	Cubic Feet.	Lbs.
ins.					ins.				
1·0	27·19	11·71	17·95	0·152	9·0	3826	1649	2525	21·40
1·5	69·51	30·00	45·94	0·389	10·0	4852	2100	3202	27·14
2·0	129·7	54·12	75·60	0·646	12·0	7300	3144	4818	40·83
2·5	213·9	92·12	141·2	1·196	14·0	10,357	4012	6836	57·91
3·0	322·6	138·9	212·9	1·804	16·0	13,950	6000	9207	78·00
3·5	456·3	196·6	301·2	2·552	18·0	18,018	7760	12,000	101·7
4·0	616·8	265·6	407·7	3·456	20·0	23,050	9930	15,220	129·0
5·0	1019	438·9	672·5	5·700	25·0	...	...	25,210	213·7
6·0	1536	661·5	1014	8·590	30·0	...	...	38,000	321·8
7·0	2180	939·0	1419	12·02	35·0	...	...	53,600	453·6
8·0	2938	1263	1939	16·43	40·0	...	...	71,800	608·5

N.B.—For other pressures—*e.g.* 150 lbs. absolute, multiply by 0·75; 180 lbs., by 0·9; 215 lbs., by 1·075; 225 lbs., by 1·125; 240 lbs., by 1·2.

For the flow of superheated steam, *vide* Table LXXVa., p. 261.

Table CXVII.—Pounds of Steam delivered per hour through Pipes with loss of 1 lb. per 100 feet.

Bore of Pipe.	Pressure of steam absolute in lbs. per square inch.												
	2	5	10	20	50	75	100	125	150	175	200	225	250
ins.													
1	...	...	...	..	104	123	144	160	174	187	200	212	222
2	...	...	...	382	590	718	815	903	984	1,060	1,130	1,198	1,265
3	356	549	760	1,054	1,620	1,975	2,245	2,490	2,715	2,925	3,120	3,310	3,450
4	750	1,140	1,550	2,085	3,120	3,850	4,500	5,060	5,600	6,100	6,600	7,050	7,500
5	1,300	1,970	2,700	3,610	5,115	6,650	7,900	8,960	9,910	10,720	11,450	12,100	12,700
6	2,015	3,100	4,290	5,960	9,170	11,000	12,700	14,000	15,350	16,500	17,640	18,600	19,500
7	3,010	4,580	6,320	8,800	13,600	16,500	19,000	21,050	22,900	24,500	26,000	27,400	28,700
8	4,200	6,730	8,830	12,300	19,200	23,050	26,300	29,100	31,800	34,050	36,150	38,200	40,100
9	5,550	8,550	11,820	16.400	25,300	30,500	35,000	38,900	42,300	45,500	48,600	51,300	54,000
12	11,380	17,500	24,300	33,700	51,800	62,800	71,700	79,500	86,600	93,300	99,600	106,000	112,000
15	19,900	30,610	42,400	58,900	90,500	...	...	...	...	...	...	...	...
18	31,400	48,400	67,000	93,000	...	...	...	...	...	...	...	...	...
21	46,100	68,300	99,500	...	...	...	...	...	...	...	...	...	...
24	64,400	90,200	137,000	...	...	...	...	...	...	...	...	...	...
27	85,000	114,000	...	...	...	...	...	...	...	...	...	...	...
30	105,000	130,000	...	...	...	...	...	...	...	...	...	...	...

N. B.—This table is based on the assumption that the only resistance to flow arises from the friction of the projecting bends; valves, branches, and other such things found in practice add to the resistance and therefore cause a diminution of flow, or a greater reduction in pressure for the same flow.

Stop-valves.—The diameter of the boiler stop-valves is often fixed from the previously determined size of main steam pipe at engines. When the branch pipes from two or more separate but similar boilers join together into one main steam pipe, of diameter D, the size of each branch pipe (their number being n) may be that given by,—

Rule 275.—Diameter of Branch pipe to each boiler $= D\sqrt{\frac{4}{3n}}$.

The more important points connected with the design and construction of stop-valves have already been dealt with (pages 286 and 392), and it is only necessary to add here that boiler stop-valves should have turned spigots fitting accurately into the holes in the boiler plates,—which should be carefully cut out by means of a bar and cutter, and *not* by hammer and chisel,—and that the flanges and necks should be extra strong and well ribbed, and the bolts or studs (bolts with heads inside the boiler are best) by which they are attached to the boiler, larger and more numerous than in pipe flanges of similar size: a §-inch bolt should be the smallest size used for attaching boiler mountings,—even for the smallest valve or cock.

In Naval work the boiler stop-valves are generally made entirely of bronze, and are of the non-return type: non-return valves are also placed at the various bulkheads, in order to localise as far as possible the effects of injury to boilers or pipes by shot, &c.

Internal steam pipes.—Where the steam room in a boiler is small relatively to the I.H.P. derived from the boiler, internal pipes with closed ends, and provided with sufficient narrow transverse slits (saw-cuts), or small holes, to give a clear area equal at least to twice that of the pipe section, should be fitted. They are best made of sheet brass —*not* copper. The number and arrangement of the pipes must be determined in accordance with the conditions of the case, but a good plan is to fit two pipes running the full length of the boiler; so that, with the stop-valve at one end, there are two pipes leading to it, each rather more than half the sectional area of the valve or of the branch steam pipe from the boiler, and with the stop-valve on the shell at mid-length there are four shorter pipes converging to it.

The steam is then gently collected from a large area, and strong currents, which might induce priming, are avoided.

Safety-valves.—The size of safety-valve should be such that it is capable of discharging all the steam that can be generated in the boiler, without allowing the pressure to rise more than 10 per cent. above that to which the valve is nominally loaded; it therefore depends mainly on quantity of fuel burnt per hour and the working pressure. A convenient and easily applied rule is,—

Rule 275a. Area in square inches of each of two valves =

$$\left(\frac{\text{Grate area}}{20}+\frac{\text{Heating surface}}{200}\right)\times\sqrt{\frac{100}{\text{Working pressure}}}.$$

Minimum Spaces between the Smoke Tubes of Small Horizontal Boilers not exceeding 10 Feet in Diameter in 32nds of an Inch, as recommended by B.M.E.D. & C. Committee.

External Diameter of Tubes.	Length of Tube between Plates in Feet.							
	5·0	5·5	6·0	6·5	7·0	7·5	8·0	8·5
1·5 inches and under . .	22	23	24	25				
1·75 inches . . .	24	25	26	27	28			
2·0 and 2·25 inches . .	26	27	28	29	30	31		
2·5 ,, 2·75 ,, . .	28	29	30	31	32	33	34	
3·0 ,, 3·25 ,, . .	30	31	32	33	34	35	36	37

SAFETY-VALVES, &c.

Safety-valves.—The Board of Trade and the Register Societies determine their size as follows:—

Rule 276. Aggregate area in square inches = total heating surface in sq. ft. $\times \frac{K}{p+15}$,

where p is the working pressure.

Tank boilers, K is 1·25 for coal-fired boilers and 1·5 for oil-fired ones.

Water-tube boilers, K is 1·0 for coal-fired boilers and 1·25 for oil-fired ones.

There must be at least two valves to each boiler. Spring valves must be cased in so as not to permit of overloading, but can be lifted by means of gear and turned round on their seats by hand.

The British Marine Engineering Design and Construction Committee recommended that the load on any single valve should not exceed 2,600 lbs.

Two or more valves may be in one chest, which must be connected to the boiler by a strong neck, the area through which need not be more than one-half the aggregate area of the valves; that is, if there are three valves each 3 inches diameter, the diameter inside the neck need not exceed 3·67 inches.

Rule 276a. The waste steam pipe and passage leading to it need have an area of cross-section no more than 0·01 × total heating surface

in sq. ft., nor less than 1·1 × the combined areas of the valves. That is, with the above three valve box, the waste steam pipe may be 5·5 inches in diameter.

During a test of 15 minutes with stop valves closed and under full firing conditions, the accumulation must not exceed 10 per cent. of W.P.

All superheaters must have an independent safety-valve at or connected to the delivery end; the diameter need not exceed $1\frac{3}{4}$ inches. Means of draining the superheaters must also be provided.

Rule 278 (French Govt.). Diameter of valve if one only $= 1{\cdot}23\sqrt{\frac{\text{T.H.S.}}{p+9}}$. (In English measures.)

Rule 279 (German Govt.). The clear area through valve seats to be so many square millimetres per square metre of total heating surface, the range is from 131 for a pressure of 75 lbs. to 51 for 240 lbs.

Wherever possible, the valves should be placed with the spindles vertical, as the action of the valve (which is very sensitive to, and easily affected by, any increase in the friction of its various parts) is then more certain. For the same reason all the parts should be a very easy fit, and the rubbing surfaces of the spindle should be draw-filed and polished with emery cloth in the same direction.

The whole of the parts must be cleaned from time to time, when in use, and the greatest care must be taken to avoid making any dints or burrs on the working surfaces.

The spring should be so proportioned that its initial compression at load is not less than half the diameter of the valve; this result can be obtained with various proportions of spring, and when height is limited a shorter spring of larger diameter and section of steel may be used.

In Naval practice, the valves, valve-boxes, &c., are usually made entirely of bronze, and the valve is generally made as a separate piece (not cast with the spindle), while the casing gear, when pressure is moderate, may be fitted to lift the valve from below. This construction has the advantage of preventing any bending or springing of the spindle that may arise from the ends of the spring not being quite true and square by its axis, from affecting the tightness of the valve on its seat; and also permits examination of the valves to be made without disconnecting and taking down easing gear shafts, &c.

The valve faces should always be flat, *not* angled at all.

When the design of the valve permits, it is a great convenience to those who will afterwards have charge of the machinery, to have a thread cut on the upper end of the spindle, so that, before taking the valve to pieces for examination or re-grinding, a nut may be put on to prevent the release or expansion of the spring when the joint at the base of the spring tower is broken; spring, tower, spindle and valve can then be readily and quickly removed in one piece.

When loose valve seats are used (as with cast-iron and steel chests) they should be securely fixed in place by a flange, or lugs, with studs or screws.

The valves, seats, spindles, compressing screws and nuts, spring washers, spindle bushes and cotters, studs and nuts for valve seats, and bushes for bearings of easing gear shafts, should be of special bronze suitable for high temperatures.

The valve may be with or without lip, but valves without lip are much less violent in action.

The point of the compressing screw should be well rounded and should enter from $\frac{3}{8}$ to $\frac{1}{2}$ inch into the spring washer.

Spiral springs.—The size of steel required by the Board of Trade is given by,—

Rule 280.
$$d = \sqrt[3]{\frac{S \times D}{C}},$$

where d = diameter or side of square of steel *in inches* (min. $\frac{1}{4}$ inch).
D = diameter of coil (centre to centre of wire) in inches.
S = load on spring in pounds.
C = 8000 for round and 11,000 for square steel.

In Naval work the values adopted for C are commonly 11,000 and 15,000 for round and square steel respectively.

d should, as recommended by the late T. W. Traill, always equal $\frac{D}{5}$: when this proportion of spring is adopted the above rule becomes,—

$$\sqrt{\frac{S}{1600}} = d \text{; or } 1600d^2 = S \text{ for round steel;}$$

$$\sqrt{\frac{S}{2200}} = d \text{; or } 2200d^2 = S \text{ for square steel.}$$

The relation between load on, and compression of, spiral springs is given by,—

Rule 281.
$$\text{Compression in inches} = \frac{D^3 \times S \times N}{d^4 \times a},$$

where N = number of free coils (not counting those which are in contact);
d = diameter or side of square of wire in *sixteenths of an inch*;
a = 26 for round, 32 for square steel;

and the other symbols have the same meanings as above.

Table CXVIII.—Safety-Valve Springs by Board of Trade Rules and the above formula.

Dimensions in inches.		Working loads in lbs.		Dimensions in inches.		Working loads in lbs.	
Diameter or side of square of steel.	Mean diameter of coil.	Round steel.	Square steel.	Diameter or side of square of steel.	Mean diameter of coil.	Round steel.	Square steel.
¼	1¼	100	137		2 5/32	3 29/32	976	1342
9/32	1 13/32	126	174	1 3/16		4 1/16	1056	1452
5/16	1 9/16	156	214		2 7/32	4 7/32	1139	1566
11/32	1 23/32	189	260	⅞		4 3/8	1225	1684
⅜	1⅞	225	309		2 9/32	4 17/32	1314	1806
13/32	2 1/32	264	363	1 5/16		4 11/16	1406	1933
7/16	2 5/16	306	421		2 11/32	4 27/32	1501	2064
15/32	2 11/32	351	483	1		5	1600	2200
½	2½	400	550		1 3/32	5 5/32	1701	2339
17/32	2 31/32	451	620	1 1/16		5 5/16	1806	2483
9/16	2 15/16	506	696		1 5/32	5 15/32	1914	2631
19/32	2 31/32	564	775	1⅛		5⅝	2025	2784
⅝	3⅛	625	859		1 7/32	5 25/32	2139	2941
21/32	3 5/32	689	947	1 3/16		5 16/16	2256	3102
11/16	3 7/16	756	1039		1 9/32	6 5/32	2376	3267
23/32	3 19/32	826	1136	1¼		6¼	2500	3437
¾	3¾	900	1237	...	...	...	...	...

The late Mr Traill fixed the values of a at 22 and 30 from experiments he had made, whilst Rankine gave 20 to 23 for round and 28·6 to 32·9 for square steel.

The superior transverse elasticity of the round sections, and especially of small ones, compared with square ones, is probably due to the fact that the round wires are often drawn, whilst the square ones are only rolled; also the material of square section steel is probably more affected by the coiling process than is that of the round.

Feed check valves.—Each boiler must have at least two independent means of feed, each with its own feed check valve. There should be interposed between them and the boiler a shut-off cock or valve to permit of their examination when necessary. The internal diameter of their seats should be at least ¼-inch more than that of the feed pipes.

Rule 282. Area through main feed valve in square inches = total heating surface in sq. feet ÷ 250; or

Minimum area of pipe section = water supplied in lbs. per hour ÷ 2500;

say, Diameter of main feed pipe $= \frac{\sqrt{\text{pounds water per hour}}}{45}$;

Rule 282a.

and, Diameter of auxiliary feed $= \frac{\sqrt{\text{pounds water per hour}}}{49}$.

or, Area of section of auxiliary feed = T.H.S. sq. ft. ÷ 300.

Feed check valves should be very strongly made, and entirely of the best bronze; the necks by which they are attached to the boilers should be specially strong, and well ribbed to the flanges. The spigots should be turned, and the holes in the boiler plates carefully bored and faced to suit. The working faces of the valves should be flat, and proportioned in accordance with Rule 153 (p. 169). The spindles should be very stout, and provided with square threaded screws; if space allows the nuts to be placed in external crossheads or bridges, they are better so arranged.

The Admiralty require that the main feed check valve shall always be placed at the right hand, as one faces the boiler and the auxiliary valve at the left hand.

Internal feed pipes.—The feed water should be led by an internal pipe (of brass) placed 2 or 3 inches below the water level, to a part of the boiler where there is a descending current, and there delivered downwards through a number of fine transverse slits or small holes, Care must be taken that this internal pipe is arranged to be kept always filled with water, and not partly with steam, so as to cause a severe water hammering action to be set up, the joints started, and the pipe very quickly destroyed.

Blow-off and scum valves.—These should also be stoutly made valves of bronze,—ribbed as above directed for feed check valves,—and are best fitted so that the pressure tends to hold the valve on its seat, in order to reduce the risk of leakage as much as possible. Now that fresh water is so much used, and loss made up by means of evaporators, and from reserve tanks, the blow-off valve is not so necessary as formerly, and is now often omitted, the boilers being emptied into the bilges when cold.

The scum valve should have an internal pipe leading to it from a circular "scum pan" or dished plate of sheet brass (about 15 inches diameter) fixed near to the centre of the water surface, and at about the lowest working level; or, if preferred, the internal pipe may reach a little beyond the centre of water surface, and may be closed at the end, and have simply a few longitudinal slits in its upper side to admit 'm.

where ar area through the blow-off valve may be that given by,—

$$d = \frac{\text{Area of blow-off pipe in square inches}}{1 + \frac{\text{Tons of water in boiler}}{25}}.$$

and i in diameter per foot diameter of boiler will modern conditions.
may be one-third that of blow-off pipe.

Water-gauge.—Every boiler must have at least two independent means of indicating the water level.

The glass tubes are $\frac{3}{4}$ or $\frac{5}{8}$ inch external diameter, and their standard lengths 20, 18, 16 or 14 inches.

All single-ended boilers over 16 feet diameter shall have a glass gauge on each side; under 16 feet one glass gauge on one side, and a set of test cocks near the other.

Double-ended boilers shall have a glass gauge near each end on opposite sides, and a set of cocks at each end.

Test cocks should be fitted direct on the boiler shell, and two in number up to 7·5 feet shells; over that there shall be three. The stand pillars must be $2\frac{1}{2}$ inches bore with $1\frac{1}{2}$-inch pipes connecting them to the boiler.

Table CXIX.—Weight of Pure Water at Different Temperatures.

Temperature of Water.	Absolute Pressure of Steam.	Weight in Lbs.		Temperature of Water.	Absolute Pressure of Steam.	Weight in Lbs.	
		A Cub. Ft.	A Gallon.			A Cub. Ft.	A Gallon.
F.°	lbs.			F.°	lbs.		
60	0·2	62·37	10·0025	303	70	57·16	9·166
90	0·7	62·13	9·964	312	80	56·85	9·117
120	1·7	61·74	9·901	320	90	56·57	9·072
136	2·7	61·45	9·858	327	100	56·33	9·035
150	3·7	61·18	9·812	334	110	56·07	8·991
160	4·7	60·98	9·780	341	120	55·83	8·953
168	5·7	60·81	9·752	347	130	55·62	8·920
175	6·7	60·66	9·728	353	140	55·41	8·885
181	7·7	60·53	9·707	358	150	55·21	8·855
187	8·7	60·39	9·685	363	160	55·02	8·824
192	9·7	60·27	9·666	368	170	54·84	8·795
196	10·7	60·17	9·650	373	180	54·67	8·766
201	11·7	60·05	9·630	377	190	54·50	8·741
205	12·7	59·95	9·614	381	200	54·34	8·717
209	13·7	59·84	9·596	385	210	54·20	8·693
212	14·7	59·76	9·584	389	220	54·03	8·668
215	15·7	59·74	9·581	393	230	53·87	8·650
228	20	59·43	9·532	397	240	53·75	8·619
240	25	59·10	9·479	400	250	53·63	8·590
250	30	58·75	9·430	404	260	53·46	8·574
259	35	58·54	9·389	407	270	53·39	8·558
267	40	58·30	9·349	410	280	53·34	8·548
275	45	58·04	9·312	413	290	53·32	8·540
281	50	57·86	9·300	416	300	53·31	8·538
293	60	57·50	9·220				

If cocks or valves are to be fitted where these connections join the boiler, they should be in sight and not capable of being closed without advertising the fact,—as serious accidents have occurred through their use,—and care should be taken not to place the inlets near where a current of steam or water may be, since the level of the water in the glass may be affected thereby,—a difference of pressure of one-tenth of a pound causing an alteration in level of 2·7 inches.

Also, in fixing the level at which the water is to stand in the glass, the lowering of level due to the cooling and consequent contraction of the water in the stand-pipe connection should not be forgotten,—though it is always an error on the side of safety.

Rankine gave the following rule for determining the volume of water at different temperatures:—

Let the volume of the water at its temperature of maximum density (39·2° F.) be represented by unity, and let its volume at T° be V, then,—

Rule 284. $$V = \frac{1}{2}\left(\frac{T+461}{500} + \frac{500}{T+461}\right).$$

He stated that the error is only one four-hundredth.

Circulating apparatus.—It is very desirable that large rigidly constructed boilers should be fitted with an apparatus for circulating the water whilst getting up steam; if hydrokineters are not supplied, connections should be made with one of the auxiliary steam pumps.

In almost all cylindrical boilers the rate of evaporation may be improved by fitting a proper arrangement of circulating plates; the first cost is small, but they are rather in the way when the boiler is being cleaned; the gain, however, is often worth the cost and trouble in crowded boilers which have to be forced.

An air-cock should be fitted at the highest point of each boiler.

THE BOARD OF TRADE RULES FOR BOILER MOUNTINGS, &c.

185. No boiler or steam chamber should be so constructed, fitted, or arranged that the escape of steam from it through the safety-valves required by the Act of Parliament can be wholly, or partially intercepted by the action of another valve.

A stop-valve should always be fitted between the boiler and the steam pipe, and, where two or more boilers are connected with a steam receiver or superheated, between each boiler and the superheater or steam receiver.* The necks of stop-valves should be as short as practicable and the chests should be tested when new to double the water pressure.

186. **Water-gauges, test cocks, and gauges.**—Each boiler should be fitted with a glass water-gauge, at least three test cocks, and a steam gauge. Boilers that are fired from both ends, and those of unusual width, should have a glass water-gauge and three test cocks at each end

* The object of this is obvious, viz. to avoid the failure of all the boilers through the failure of one.

or side, as the case may be. An additional glass water-gauge may, however, be substituted for three test cocks. When a steamer has more than one boiler, each boiler should be treated as a separate one, and have all the requisite fittings.

When the water-gauge cocks are not attached directly to the shell of the boiler, but to a stand-pipe or column, cocks should as a general rule be fitted between the boiler and the stand-pipes, &c., and may be placed either on the boiler or at the stand-pipe. Such cocks need not, however, be insisted on in cases where the columns, stand-pipes, &c., are of moderate length and of suitable strength, provided that the diameter of the bore of any part is not less than 3 inches.

Valves between boiler and stand-pipe should not be passed.

If the column, stand-pipes, &c., are of less diameter than 3 inches, and the pipes are bolted to the boiler without the intervention of cocks, the arrangement need not be objected to, if otherwise satisfactory, providing there is no difficulty in keeping the passages at the ends clear, and ascertaining that they are so. To do this it will be necessary that the passage in the part of the column between the top and bottom gauge-glass cocks be cut off or closed, which may be done permanently, or by the interposition of a cock at that part. The latter is a convenient and desirable arrangement even when cocks are fitted on the boiler.

In the case of high-pressure boilers, it is desirable that the cocks in connection with the water-gauges should be fitted with handles which can be expeditiously manipulated from a convenient position.

It is desirable in all cases that test cocks should be fitted directly to the skin of the boiler; and when the water-gauge is attached to a column, the opening through which is stopped or can be cut off, the test cocks *must* be fitted directly to the skin of the boiler.

The Surveyors should satisfy themselves by actual examination whether the glass water-gauges of the boilers of the vessels they survey are clear, and also whether they are fitted with automatic valves or fittings, as the existence of such fittings cannot always be ascertained by external examination. In all cases where automatic gauges are fitted, full particulars thereof should be submitted for consideration and approval before the gauges are passed.

191. Cast-iron stand-pipes or cocks intended for the passage through them of hot brine should not be passed.

Surveyors should also discourage the use of cast-iron chocks and saddles for boilers, and particular attention should be paid to the chocking of boilers, more especially when they are fired athwartships.

The Board of Trade Rules for Safety-Valves are as follows:—

177. Cases have come under the notice of the Board of Trade in which there were pipes between the boilers and the safety-valves. Such arrangement is not in accordance with the Act, which distinctly provides that the safety-valves shall be upon the boilers.

The Surveyors are instructed that in all *new boilers*, and whenever *alterations can be easily made*, the valve chest should be placed directly

on the boiler; and the neck, or part between the chest and the flange which is bolted on to the boiler, should be as short as possible, and be cast in one with the chest.

The Surveyors should note that it is not intended by this instruction that vessels with old boilers which have been previously passed with such an arrangement should be detained for the alterations to be carried out.

Of course, in any case in which a Surveyor is of opinion that it is positively dangerous to have a length of pipe between the boilers and the safety-valve chest, it is his duty at once to insist on the requisite alterations being made before granting a declaration.

If any person place an undue weight on the safety-valve of *any* steamship, or, in the case of steamships surveyed under the Act, increase such weight beyond the limits fixed by the Engineer Surveyor, he shall, in addition to any other liability he may incur by so doing, be liable for each offence to a fine not exceeding one hundred pounds.

179. The locked-up valves, *i.e.* those out of the control of the engineer when steam is up, should have an area not less, and a pressure not greater, than those which are not locked up, if any such valves are fitted.

When natural draught is used, the area per square foot of T.H. surface of the locked-up safety-valves should not be less than that given by the Rule (page 394) for the boiler pressure intended, but in no case should the valves be less than 2 inches in diameter. This applies to new vessels or vessels which have not received a passenger certificate.

When, however, the valves are of the common description, and are made in accordance with the Rule, it will be necessary to fit them with springs having great elasticity, or to provide other means to keep the accumulation within moderate limits.

The Board of Trade will now permit the use of some other equally good and reliable safety-valve in lieu of the valve of ordinary type, and if satisfactory, allow a reduction in area of as much as 33 per cent.

When the pressure exceeds 180 lbs. per square inch the accumulation of pressure at the steam test will probably be exceptionally high, unless the area of the branch leading from the valve chest is in excess of the area of the valves, and the area of the main waste steam pipe is correspondingly in excess of the gross area of the valves.

In ascertaining the fire-grate area, the length of the grate should be measured from the inner edge of the dead plate to the front of the bridge, and the width from side to side of the furnace on the top of the bars at the middle of their length.

In the case of vessels that have not had a passenger certificate, if there is only one safety-valve on any boiler, the Surveyor should not grant a declaration without first referring the case to the Board for special instructions.

Table CXX.—Safety-Valve Areas for Cylindrical Boilers Using Coal Fuel.

Boiler Pressure.	Aggregate Area of Valves per 100 sq. ft. of Total Heating Surface.	Boiler Pressure.	Aggregate Area of Valves per 100 sq. ft. of Total Heating Surface.	Boiler Pressure.	Aggregate Area of Valves per 100 sq. ft. of Total Heating Surface.	Boiler Pressure.	Aggregate Area of Valves per 100 sq. ft. of Total Heating Surface.
120	0·926	165	0·700	210	0·556	255	0·463
125	0·893	170	0·676	215	0·544	260	0·455
130	0·862	175	0·658	220	0·532	265	0·447
135	0·833	180	0·642	225	0·521	270	0·439
140	0·806	185	0·625	230	0·511	275	0·431
145	0·781	190	0·610	235	0·500	280	0·424
150	0·758	195	0·595	240	0·491	285	0·417
155	0·736	200	0·582	245	0·481	290	0·410
160	0·715	205	0·570	250	0·472	295	0·404

For oil-fired boilers multiply by 1·2 or divide by 0·833. For the area through the neck take one-half these.

For the area through cross-section of waste steam pipe and passages thereto, the allowance is 1 square inch per 100 square feet of total heating surface in each case.

180. The safety-valves should be fitted with lifting-gear, so arranged that the two or more valves on any one boiler can at all times be eased together, without interfering with the valves on any other boiler. The lifting-gear should in all cases be so arranged that it can be worked by hand either from the engine-room or stoke-hold.

Care should be taken that the safety-valves have a lift equal to one-fourth of their diameter; that the openings for the passage of steam to and from the valves, including the waste-steam pipe, have each an area not smaller than the area required by Section 179, and the area of the main waste-steam pipe should not be smaller than the combined area of the branch pipes. Each valve box should have a drain pipe fitted at its lowest part.

Too much care cannot be devoted to seeing that there is proper lift, also that free means of escape for the waste steam are provided, as it i

obvious that, unless the means for escape of the waste steam are ample, the effect is the same as reducing the area of the valves or putting an extra load upon them. The valve seats should be secured by studs and nuts.

181. The following conditions are to be complied with :—

(1) That at least two valves are fitted to each boiler.
(2) That the valves are of the proper size, as by clause 179.
(3) That the springs and valves are so cased in that they cannot be tampered with.
(4) That provision is made to prevent the valves flying off in case of the springs breaking.
(5) That screw lifting-gear is provided to ease all the valves, as by clause 180.
(6) That the size of the steel of which the springs are made is in accordance with that found by the following formula :—

$$\sqrt[3]{\frac{s \times D}{c}} = d :$$

s = the load on the spring in lbs.
D = the diameter of the spring (from centre to centre of wire) in inches.
d = the diameter, or side of square, of the wire in inches.
c = 8000 for round steel.
c = 11,000 for square steel.

(7) That the springs are protected from the steam and impurities issuing from the valves.
(8) That when valves are loaded by direct springs, the compressing screws abut against metal stops or washers, when the loads sanctioned by the Surveyor are on the valves.
(9) That the springs have a sufficient number of coils to allow a compression under the working load of at least one quarter the diameter of the valve.

The size of steel of springs of safety-valves should not as a rule be less than $\frac{1}{4}$ inch.

182. **Safety-valve steam tests.**—In no case is the Surveyor to give a declaration for spring-loaded valves, unless he has tried them under full steam, and full firing, for at least 15 minutes with the feed-water shut off and stop-valve closed, and is fully satisfied with the result of the test. If the accumulation of pressure exceed 10 per cent. of the loaded pressure he should withhold his declaration and report the case to the Board of Trade.

183. The tracings of new safety-valve designs should, if possible, be transmitted to the Board of Trade for consideration before the construction of the safety-valves is commenced.

178. **Liability of owners, &c.**—It is clearly the duty of the

masters and engineers of vessels to see, in the intervals between the surveys, that the locked-up safety-valves, as well as the other safety-valves, and the rest of the machinery, are in proper working order. There is no provision in the Merchant Shipping Act, 1894, exempting the owner of any vessel, on the ground that she has been surveyed by the Board of Trade Surveyors, from any liability, civil or criminal, to which he would otherwise be subject. The Act of Parliament requires the Government safety-valves to be out of the control of the engineer when the steam is up; this enactment, far from implying that he is not to have access to them, and to see to their working, at proper intervals when the vessel is in port, rather implies the contrary; and the master should take care that the engineer has access to them for that purpose. Substantial locks that cannot be easily tampered with, and as far as possible weather-proof, should be used for locking up the safety-valve boxes.

In witnessing the tests of boilers, &c., and safety-valves, the Surveyors are to use the pressure gauges supplied by the Board of Trade. The steam gauge should not be used without a syphon filled with water between it and the boiler.

ADMIRALTY RULES FOR SAFETY-VALVES.

221. **Safety-valves.**—Each boiler is to be fitted with at least two vertical safety-valves of approved design, loaded to 235 lbs. per square inch, with springs on an approved plan, and placed on the boilers independently of the internal steam pipes.

The spindles are to have a suitable joint outside the valve box to enable the valves to freely adjust themselves on their seatings.

The safety-valves are to be so adjusted that they will commence to lift at the specified working pressure, and be fully open with an increased pressure of not more than 7 lbs. The valves must also close on their seats when the pressure falls to the specified working pressure, and be quite tight at not more than 7 lbs. below this pressure.

222. **Safety-valve area.**—The total clear disc area of the safety-valves on each boiler, at the most restricted part when just lifted, is to be not less than $5\frac{1}{4}$ square inches per 1000 square feet of tube heating surface for ordinary type valves, and as may be approved for valves of special design.

223. **Accumulation test.**—After these valves have been adjusted under steam, and the stops have been fitted to ensure the valves lifting at the specified working pressure, a further trial of at least 30 minutes duration is to be made with one boiler, with the stop-valves closed, and

burning oil fuel at a rate corresponding to not less than 1 lb. per square foot of heating surface.

During these trials the accumulation of pressure, as registered by the pressure-gauges on the boiler, is to be not greater than 7 per cent. of the specified working pressure.

This trial is to be carried out prior to the trials mentioned in Clause 4.

224. **Safety-valve lifting gear.**—Gear with suitable indexes is to be fitted to enable the safety-valves to be easily lifted from the boiler-room floors at readily accessible positions, and also from suitable and accessible positions on deck. Cast steel levers or wire ropes are not to be used, and no part of the lifting-gear is to be fitted inside the valve boxes. The lifting-shafts and cams are to be forged from the solid. All the joints of the safety-valve lifting gear are to be bushed with gun-metal, and provision is to be made for convenient access to the various parts, and for efficient lubrication. The lifting gear is to be strong enough to lift the valves easily under all conditions, and is to be so designed that the valves are opened with a left-hand motion of the wheel.

225. **Safety-valve springs.**—The safety-valve springs are to be fitted with guides at the top and bottom, and also, if required, at the middle; the dimensions of the steel of the springs are to be such that the amount of compression when screwed down to the maximum working load shall not be less than the diameter of the valve, with sufficient clearance to admit of further compression of one quarter the diameter of the valve. Provision is to be made to prevent the valves being blown out of their seats in the event of the springs breaking.

226. **Fittings for water testing.**—All the necessary fittings are to be supplied for gagging the safety-valves for the water tests of the boilers.

BOILER MOUNTINGS AND FITTINGS, B.M.E.D. & C. COMMITTEE.

Water gauges, two to each boiler. Tubes ¾ inch or ⅝ inch outside diameter, and 20, 18, 16, or 14 inches long. Double-ended boilers over 16 feet diameter to have three. Stand pipes at least 2½ inches diameter inside.

Pressure gauge, one to single-ended, two to double-ended boilers.

Salinometer cock, one fitted direct to each boiler.

Check feed valves, two independent valves to each boiler, with shut-off cocks or valves between them and the boiler.

Blow-off valve to each boiler not more than 1½ inches diameter.

Stop-valves for steam, one main, one auxiliary to each.

Valves for steam steering and whistles; and when more than one boiler, two at least shall have these valves.

All valves over 1½ inches diameter must have outside screws, and shut with right-hand motion.

All cocks and valves shall be such as to be seen whether open or shut.

Stop and safety valves, chests for temperatures over 425° F. must be of cast steel.

Feed valves, chests, and all cocks and valves should be of strong tough bronze, and those exposed to steam should be such as to maintain good strength and toughness while at the steam temperatures.

All chests of stop-valves, safety-valves, check-feed valves, and the shells of cocks over one inch diameter shall be tested to twice the working pressure.

LLOYD'S RULES FOR BOILER MOUNTINGS, &c.

Generally Lloyd's Register and the other Societies adopt the Rules recommended by the B.M.E.D. & C. Committee and as accepted by the Board of Trade as set out on pages 401, etc. It was Lloyd's rule, however, to allow the safety-valves to be loaded to 5 lbs. in excess of the working pressure when it did not exceed 60 lbs. per square inch. The Committee recommended that all safety-valves should be so set that one on each boiler should blow at a pressure 2½ per cent. in excess of working pressure and not more than 5 lbs., and the remainder at 3½ per cent., or not more than 7 lbs., which is reasonable and better than blowing off with great noise and loss of steam at every casual stoppage or with the running working pressure kept considerably below the working pressure for which the boiler is made.

If improved valves are used, they are to be tested under steam in the presence of the Surveyor; the accumulation is in no case to exceed 10 per cent. of the working pressure.

An approved safety-valve should be fitted to every superheater near or at the delivery end, so that when the engine is not taking steam, there may be passage of it through the superheater. This valve should be set to blow slightly below that at which the main safety-valves blow.

In winch boilers one safety-valve will be allowed of the size given by rules; the Board of Trade require two valves in all boilers.

8. Each valve is to be arranged so that no extra load can be added when steam is up, and must be fitted with easing gear which must lift the valve itself. All safety-valve spindles are to extend through the covers, and are to be fitted with sockets and cross handles, allowing them to be lifted and turned round in their seats, and their efficiency tested at any time.

9. Stop valves are to be fitted so that each boiler can be worked separately.

10. Each boiler is to be fitted with a separate steam gauge, to accurately indicate the pressure.

11. Each boiler is to be fitted with a blow-off cock, independent of that on the vessel's outside plating.

FURNACE FITTINGS.

Furnace fronts, &c.—Furnace fronts and fire doors may with advantage be of wrought-iron or steel,—although they are often of cast-iron; but the internal protecting or baffle plates are better of cast-iron, since it burns away less rapidly than the wrought material, and should be in several small pieces, free to expand in all directions, rather than in one large piece, which would probably very soon crack and get adrift.

The size of fire door, in the clear, may vary from 12 inches high by 18 inches wide, to 16 inches high by 24 inches wide,—the top of the opening being arched, and struck with a radius equal to the *width* of the opening.

A very good arrangement is that in which the otherwise useless corners to right and left of, and above the fire door, inside the furnace, are filled in by curved cast-iron plates, perforated with small holes to allow the air to enter the furnace; the air is thus heated to a certain extent in these boxes or chambers before coming into contact with the fuel, and the riveted joint round the furnace mouth is protected from the fire.

Furnaces over 3 feet 6 inches diameter are perhaps better with two half-doors hinged right and left, and meeting in the centre, only one of which need be opened at once; less cold air is then admitted, and one side of the fire can be attended to at a time.

Means should be provided for holding the doors open in a sea-way.

Fire-bars, &c.—When chimney draught only, or chimney and forced draught not exceeding ·5 inches of water, is used, the bars are better in one length,—up to 5 feet 6 inches long; when a greater air pressure is used, or when the grate is longer than 5 feet 6 inches, two lengths of bars may be used.

When one long bar is used, it should be hooked to the inner edge

of the dead plate, and free to expand inwards, and slide on the bridge-plate; when two lengths are used, they should be hooked to the central bearer-bar, and free to slide on both the bridge-plate and the dead-plate.

In either case care should be taken that the dead-plate is formed so as to hold or support the bars with their faces or upper surfaces flush with its own upper surface.

The thickness of bar and width of air space between bars must depend on the class of coal that will generally be used, and on the air pressure with which it is intended to work.

The following are good average dimensions for fire-bars:—

Thickness on face,	$^{13}/_{16}$ inch.
,, at bottom edge,	$^{3}/_{8}$,,
,, on face, over distance pieces,	$1^{3}/_{16}$,,
,, at bottom edge, ,, ,,	$1\frac{1}{8}$,,
Depth at centre,	$\cdot 66\sqrt{\text{length in inches.}}$
,, near ends,	$2\frac{1}{2}$ inch.
Width of air spaces,	$^{5}/_{8}$,,

The slope of the grate surface should never be less than 1 inch per foot of length, and is better $1\frac{1}{2}$ inches, or even more, as is possible with large ones and short bars.

When corrugated or ribbed furnaces are used, the two side bars should be made to templates from the furnaces, and should fit as closely as possible into all the recesses.

Fire-bars should, of course, be made of the most refractory iron obtainable; fine grey irons are quite unsuitable.

Bridges, &c.—In return-tube boilers the grate should never be so long that the front face of the bridge is less than 9 inches from the face of the back tube-plate.

The height of bridge should be such that the clear area above it may be from $\frac{1}{6}$th to $\frac{1}{8}$th of the area of grate; this proportion is obtained approximately when the clear height above the bridge at its centre is $\frac{1}{3}$rd of the diameter of furnace.

In the case of corrugated or grooved furnaces, the ash-pits require to be fitted with thin lining plates to enable the rake to be used. In Naval and other ships in which forced draught is used to any considerable extent, shallow pans which can be kept full of water are necessary.

Each ash-pit should also be fitted with a good stout pricker bar and with a damper; when the closed stoke-hole system of forced draught is used, the dampers are sometimes balanced and made to open inwards only,—closing against any pressure that may come from the furnace side.

LADDERS AND PLATFORMS, &c.

The following Table gives ordinary dimensions of ladders and gratings:—

Table CXXI.—Ladders and Gratings.

Ladders.					"Spill" Gratings.			
Width of ladder.	Size of side bars.		Width of cast-iron steps.	Bar steps, No., size, and pitch of bars.	Width of grating.	Size of side bars.	Diameter of "spills."	Pitch of "spills."
	Cast-iron steps.	Round bar-iron steps.						
in.	in. in.	in. in.	in.	in.	in. in.	in. in.	in.	in.
12	...	2¼ × ⅜	...	one ⅝	15 & 18	2¾ × ⅜	⅝	2¼
15	...	3¼ × ⅜	...	two ⅝, 1⅜ pitch	21	3 × 7/16	¾	2⅜
18	3½ × ⅜	3½ × ⅜	4¼	two ⅝, 1½ pitch	24	3 × 7/16	¾	2⅜
21	3¾ × 7/16	...	4¾	...	27	3 × 7/16	¾	2⅜
24	4 × 7/16	...	5¼	...	30 and above	3 × 7/16	⅝ with centre support	2¼

⅝-inch round iron "spills" should not be used for spans over 18 inches.

¾-inch round iron "spills" should not be used for spans over 27 inches.

d = Diameter of spills in sixteenths.
L = Length in inches.

Then $L = \frac{d^3}{70} + 4$.

Ladder steps should be from 9 to 10 inches apart (face to face).

The front corners of cast-iron steps should be rounded away, and they should be attached to the side bars of the ladder by two ½-inch bolts at each end.

The main engine-room ladder should be at least 21 inches wide, and in large ships, where there is plenty of room, a width of 24 to 30 inches.

Stoke-hole ladders are not usually fitted with cast-iron steps.

The inclination of a ladder to the vertical may be almost anything,

and depends on the space available and purpose for which ladder is fitted; the main engine-room ladder should be 1 in 2¼ when possible.

Handrails should be of steel or wrought-iron, 1 inch in diameter.

Stanchions, when 3 feet high, may taper from ⅞ inch diameter at the top to 1¼ inches at the bottom; when short, for ladders, ¾ inch at top to 1 inch at bottom is enough. The ball through which the rail passes may be 2 inches in diameter.

Engine and boiler-room floors are best laid with chequered wrought-iron or steel plates,—engine-room $^{5}/_{16}$ inch thick, and boiler-room ⅜ inch, exclusive of the raised pattern. In the Navy, ¼ inch and $^{5}/_{16}$ inch respectively are the usual thicknesses.

It is very desirable that floor plates should be secured to the bearers where possible, as an accumulation of water surging from side to side in the stoke-hole may lift the plates and drive them against the sea-valves and pipes, and thus cause serious damage, sufficient sometimes to imperil the safety of the ship.

ENGINE AND BOILER SEATINGS, &c.

It is extremely difficult to lay down any general rules that will be of service in designing engine and boiler seatings, because so much depends on the type and structure of ship, strength and the design of engine framing or bed-plate, type of boilers and position in which they are to be placed, &c., but the following hints may be of some use.

Seatings for vertical engines.—When the seating must be built upon the top edges of the ship's floors, it generally consists of two box girders, one under each side of the bed-plate, parallel to the shaft axis. Whether additional cross girders should be fitted under each main bearing depends on the strength of the framing or bed-plate.

Such girders should be made generally of scantling corresponding with the frames and keelsons of the ship. Their tops should be much thicker, and are usually from 1½ to 2 times the thickness of the vertical plates.

One of the chief difficulties with this type of structure is to get a sufficiently good attachment to the floors; double reverse bars should always be fitted under the engines and boilers, and for large and heavy engines the attachment of the vertical side plates to the reverse bars should also be by means of double angle bars, so that there may be four rivets at every crossing point. The second bar is sometimes represented by a separate short piece at each frame. In very lightly built ships it is often advisable to carry the vertical plates down between the floors, and to attach them to the skin plating as well as to the reverse bars.

Most ships nowadays have double bottoms fore and aft, and on them the engine girders are erected and riveted to them, thereby getting a strong attachment and a good distribution of strains. Cross girders or brackets are usually fitted for this purpose, as also to give stability to the structure and prevent racking.

In all cases where rigid girders are added to the structure of the ship for the purpose of properly distributing weights or strains, care should be taken that they do not stop abruptly at any point, bulkhead or otherwise, but gradually decrease in section, or taper down for three or four frame spaces, as otherwise serious results may ensue from the localisation of the flexure or spring of the structure.

Very large and heavy engines are generally so constructed as to require only a plain flat surface of the same length and breadth as the bed-plate for a seating, and where the ship is built with open floors this should be obtained by placing a number of longitudinal girders across the tops of the floors and plating them over, using only so many athwartship girders as may be absolutely necessary.

In either of the cases, if the shipbuilders are communicated with in time, there will not generally be any difficulty in modifying the spacing of the longitudinals slightly to suit the engines; a few inches in height may also sometimes be gained by forming troughs or recesses in the inner bottom under the cranks.

Where box girders are placed upon the inner skin as described above, the Admiralty rule is that they shall be made watertight, in order to prevent internal corrosion.

The seatings for paddle engines are usually constructed on one of the above described plans, or on a combination of them, and do not require any special description. The cases in which the framing of the engine is constructed of plates and angles and built into the ship are of too special a nature to be advantageously treated here; and the various methods of stiffening the sides of the ship and attaching the brackets for carrying the outer bearings also fall into the same category.

Thrust block seatings should have specially strong and well-extended attachments; they should extend over at least three frame spaces in small vessels, and over six or more in large ones, and some of the longitudinals of the engine seating should be made continuous with those of the thrust seat.

The rivet holes for the thrust blocks and all parts of the engine beds subject to racking should be drilled, or, say, a half of them drilled and the remainder carefully reamered out, otherwise the rivets will work slack and be inefficient.

Holding down bolts should be numerous and well distributed, and should have the ends slightly burred over to prevent the slacking back of the nuts.

Staying of engines.—The cylinders, &c., of vertical engines may be stayed, but under no circumstances rigidly to the decks or upper works of the ship; nor should the two sets of engines in a twin-screw vessel be stayed to one another; there is usually no difficulty in obtaining all necessary stiffness and stability by care in designing the framing, and the risks run by using rigid stays to the ship are sometimes serious. In paddle vessels it is not practically possible to avoid connection between the engines and the upper works of the ship, but even here, where a certain amount of spring is allowed for, cracked entablatures, or top frames, were by no means uncommon.

Boiler seatings.—When either single or double-ended boilers are placed with their axes athwartships, the best type of seating is that in which H section girders (10 or 12 inches deep) running the full length of the boiler-room are used, the lower flanges being riveted to double reverse bars, or the inner skin, and the upper flanges carry the wedge-shaped chocks which maintain the boilers in place. When the vessel is of cellular construction these girders should be arranged when possible to coincide with the longitudinals in the bottom.

When the boilers are placed with their axes fore and aft, as is now usual, each cradle or bearer is built upon the top of a separate floor; the proper distribution of the weight is more difficult, but can be effected by putting in additional intercostal longitudinals, or, preferably by fitting longitudinals on top of the floors. Care must be taken that these additional longitudinals do not prevent access to the underside of the boiler; if they are made of considerable depth, manholes may be cut in them. When the construction of the vessel is cellular, there is not usually any need to supplement the longitudinal connections.

Single-ended boilers should have two cradles or pairs of chocks; double-ended boilers of moderate size and weight, three; and very large and heavy double-ended boilers, four.

The boilers should be prevented from moving end-ways by "toe" plates or brackets riveted to some convenient portion of the seating or of the vessel's structure.

The area of section of such toe plates should be not less than $\frac{W \times K}{96}$ square inches to resist shear. W being the weight of the boiler, &c., in tons, and K the speed of the ship in knots.

The greatest care must also be taken so to secure the boilers in their seats that no possible movement of the vessel will throw any strain upon any of the pipes or connections. This is best done by riveting to the upper part of the shell of each boiler four "eye" plates, from each a rod or link is fitted leading to some convenient part of the vessel's structure (such as stringer or deck-beam), or to the neighbouring boiler. These eyes may be riveted in places when the riveting of the shell is done, and they then serve to sling the boiler by. Brackets of plate and angle-iron attached to the deck-beams and almost touching the top of the shell, acting as inverted toe plates, are also sometimes a convenient means of fixing the boilers in their seats.

The common Admiralty method of securing cylindrical boilers was to rivet four eyes to the shell at about a foot above the top of the seatings and in the same planes with them, and four other similar eyes to the tops of the seatings themselves, and then connect each pair of eyes by a pair of flat links and pins, &c.

LLOYD'S RULES FOR ENGINE AND BOILER ROOMS.

Section 30.—1. Engine and Boiler Bearers.—In steam vessels care must be taken that the engine and boiler bearers are properly constructed, and fitted with efficient longitudinal ties. Where the bearers interfere with the longitudinal strength of the vessel, they must extend a sufficient distance beyond the bulkheads of the engine and boiler space to compensate for the same.

When Engines of High Power are fitted.—Where it is intended to fit engines of greater power than in ordinary cargo-carrying steamers, the engine seating is to be of proportionately greater strength, and to be specially adapted with this object in view by being connected to the sides of the vessel. Other means are, if necessary, to be adopted in order to ensure the rigidity and strength necessary to withstand the vibration produced in this part of the vessel.

2. **Strengthening in Machinery Space.**—Additional strengthening by means of web frames, and "strong" beams, or otherwise is to be provided in the machinery space. Plans showing the proposed additional strengthening are to be submitted for approval.

3. **Clearance between Bulkheads and Boilers.**—Coal bunker bulkheads are to be kept well clear of the boilers and their uptakes. Where the boiler-room bulkhead is recessed for a donkey boiler, the recess is to be of a size sufficient to give space all round the boiler to admit of its being properly attended to.

In order to afford protection against the heat from the boiler, the roof of the recess is to be not less than 4 feet clear of the top of the boiler, the space between the bunker or hold bulkhead plating and the chimney is to be not less than 18 inches, and a baffle plate is to be fixed between the chimney and the bulkhead; other efficient means may be provided. Wood lining is to be fitted on the hold side of the recess plating with an air space between it and the plating.

4. **Tie-beams across Recess.**—When a recess extending above the hold beams is formed in the engine-room bulkheads, the bulkhead is to be efficiently connected from side to side by a tie or bridle beam at about the height of the hold beams, strongly riveted to the plating and fitted with efficient gusset plates.

5. **Protection of Deck under Donkey Boilers.**—Where vertical donkey boilers are placed on the decks of vessels, the deck underneath them is to be protected by being covered with firebrick or cement not less than 2 inches in thickness. The deck on which fires may be drawn from any donkey boiler is also to be protected by firebrick or cement not less than 2 inches in thickness.

6. **Shaft Tunnel.**—The plating of shaft tunnels is to be of the thickness required in Table 10 for the lower part of bulkhead plating; the top plating in way of the hatchways to be not less than ·10 of an inch thicker than the remaining plates, or to be covered with wood not less than 2 inches thick.

The tunnel is to be strengthened with transverse angle bars of the size of the lower deck stringer angles spaced not more than two frame spaces apart, and 3 feet in way of the hatchways.

The plating is to be caulked, and the tunnel tested with water from a hose to ensure its being watertight.

The bulkheads and top plating of tunnel recesses to be strengthened and supported by similar angles, but spaced the same as the vessel's frames; the top plating where attached to the sides of the vessel to be made watertight with steel or iron collars or chocks, to the exclusion of wood or cement.

The tunnel to be fitted with a watertight sluice door on the engine-room bulkhead, capable of being closed from the upper deck.

If a pipe tunnel is led through the forward holds, its structure is to be the same as that of a shaft tunnel.

Section 50 (Valves, etc.).—1. No sluice valve or cock is to be fitted to the collision bulkhead.

2. No sluice valves or cocks are to be fitted to the engine-room or other watertight bulkheads unless they are arranged so as to be accessible at all times.

3. If the after peak is used as a ballast tank, no sluice valve or cock is to be fitted to the after bulkhead; but if it is not so used, and if no pump is fitted in it, a sluice valve or cock is to be fitted to the after bulkhead, to allow water to reach the pumps when required.

4. When sluice valves are fitted, they are to be so arranged as to be controlled from above the load water-line, and the rods are to be boxed in to prevent injury.

5. All head and stern pumps to be provided with sea cocks fitted to the outside plating to the satisfaction of the Surveyors, and in places where they are at all times accessible.

6. Where soil pipes are attached to the outside plating below the load water-line, the lower length must be of steel or iron of substantial thickness, and be secured to the plating with a proper faced joint, and extended for some distance above the load water-line.

7. If the remainder of the pipe be of lead, care must be taken that it is of substantial thickness, and that it is properly protected externally with zinc or iron, to the satisfaction of the Society's Surveyors.

Section 31 (Openings in Decks).—1. The engine and boiler openings of the weather-deck of steam vessels are to be properly framed for a height of not less than 18 inches above the deck, the coaming plates to extend to the lower edge of the beams, and iron or steel trunk bulkheads connected to the coamings should be fitted to a height of about 7 feet above the deck.

2. The engine and boiler openings in the 'tween decks of all vessels are also to be enclosed by trunk casings efficiently stiffened by angle bars 30 inches apart, and extending to the weather-deck beams, to which they are secured.

3. Strong iron doors will be allowed in these trunk casings, provided their lower parts are at least 18 inches above the deck, and efficient arrangements made for their security.

4. When a poop, or bridge-house, covers the engine and boiler space, the coamings of the engine and boiler openings should not be less than 2 feet above such deck, unless these openings are constructed as provided for in the first paragraph of this section.

5. It is considered that in all cases the engine and boiler openings should be made as small as practicable, and be subdivided by athwartship iron divisional bulkheads to secure the maximum safety of the vessel. The two sides of the casing should in all instances be efficiently connected by angle beams within them at the upper part.

6. The engine-room skylights are to be in all cases substantially constructed and to be securely bolted or riveted to the coamings, and where the skylight top is not solid with bull's-eyes fitted in the same, efficient dead-lights of metal or wood must be provided. The grating openings over the stoke-hold must also be protected by plates, fitted with hinges, or otherwise in a manner satisfactory to the Surveyors.

7. Where either of the openings exceeds 15 feet, or the combined length exceeds 30 feet, the beams in way of the same are to be plated over from the stringer to the tie-plates, the plating extending two beam spaces beyond the openings, and tapered from thence towards the stringer plate for a distance not less than the breadth of the plating required to be fitted; the thickness of this plating to be the same as given in Table S. 5. for steel decks.

8. Where large openings are adjacent to each other, the intervening space between the openings to be plated over.

9. **Steam Trawlers.**—In all steam trawlers the deck beams should be wholly plated over in way of the engine and boiler casings, and the casings should extend down to the under side of the deck beams and be connected to the deck plating with angle bars and to the half beam ends with angle lugs. If the casings be not extended down to the under side of the beams, they should be attached to the deck plating with angles $4\frac{1}{2} \times 4\frac{1}{2} \times 0{\cdot}36$, having two rows of reeled rivets in each flange. In order to ensure that the scantlings and construction of these casings are satisfactory in every case, detailed plans of the same should be submitted for approval.

HATCHWAYS.

Section 32.—7. Connections of Coamings to Deck.—The coaming plates are to be connected to the deck plating or tie plates with angles of not less thickness than the side coaming plates, welded at the corners of the hatchways. Where a wood deck is fitted, the vertical flanges of the angle bars connecting the side and end coamings to the deck are to extend half an inch above the deck.

8. **Material.**—All hatchway coamings on weather decks and the companions at the fore-end of steamers to be of steel or iron.

9. **Half Beams.**—Where half beams are fitted to alternate frames, they are to be connected to the coaming plates with double angles, and half beams fitted to every frame may be connected to coaming plates,

with single angles of not less thickness than the side coamings. There are to be three rivets in each flange of the angles connecting coamings to the half beams where the depth of the half beam is 7½ inches to 9½ inches, and four rivets where the depth is 10 inches to 12 inches.

Section 30 (Bunker Hatches, etc.).—Coal-bunker pipes, where practicable, are to be formed so as to be at least 12 inches above the upper deck, fitted with lids having studs to fit in openings made in the pipes, for their security; the pipes to be so formed that tarpaulin may be securely lashed over them. When there are coal-bunker hatches in the weather deck, they must be properly framed with coaming plates of suitable height, having solid hatches secured by an iron bar or other approved fastening.

Section 39 (Pumps, etc.).—1. In steam vessels the pumping arrangements according to the division of holds, &c., to be as follows:—

2. **Holds with double bottoms.**—In the double bottom of each compartment of the hold, and of engine and boiler space, a steam-pump suction is to be fitted at the middle line, and one on each side to clear the tanks of water when the vessel has a heavy list. Where there is considerable rise of floor towards the ends of the vessels, the middle line suction only will be required. A steam-pump suction and a hand-pump are also to be fitted to each bilge in each hold where there is no well. When there is a well, one or three steam-pump suctions are to be fitted in the same, according as there is considerable or little rise of floor, and hand-pumps fitted at the bilges.

3. **Holds without double bottoms.**—Where there is considerable rise of floor, one steam-pump suction and one hand-pump are to be fitted in each hold. Where there is little rise of floor, two or three steam-pump suctions and at least one hand-pump are to be fitted to each hold.

4. **Engine and Boiler space.**—Where a double bottom extends the whole length of engine and boiler space, two steam-pump suctions are to be fitted to the bilge on each side. Where there is a well, one steam-pump suction should be fitted in each bilge and one in the well. Where there is no double bottom in the machinery space, centre and wing steam-pump suctions should be fitted. The rose box of the bilge injection is to be fitted where easily accessible, and is to be used for bilge water only. The main and donkey pumps to draw from all compartments, and the donkey to have also a separate bilge suction in the engine-room.

5. **Fore and After Peaks.**—If the peaks are fitted as water ballast tanks, a separate steam-pump suction is to be led to each. If not used for water ballast, an efficient pump is to be fitted to the fore peak.

6. **Tunnel.**—The tunnel well is to be filled with a steam-pump suction.

7. **All Hand-pumps** to be capable of being worked from the upper or main decks or above the deep load water-line. The pump chambers are not to be more than 24 feet above the suction rose.

The sizes of the hand-pumps are to be not less than given in following Table:—

Tonnage under upper deck.	Hand-pumps in holds.	
	Diameter of barrel.	Diameter of tail pipes.
	Inches.	Inches.
In vessels not exceeding 500 tons, . . .	4	2
Above 500 tons but not exceeding 1000 tons, .	4½	2¼
Above 1000 tons but not exceeding 2000 tons,	5	2½
Above 2000 tons, ...	5½	2¾

In lieu of hand-pumps in each compartment, an approved **fly-wheel pump** may be fitted if it is connected to the steam-pump bilge suction pipes of these compartments.

The hand-pumps may be dispensed with in vessels which have two independent boiler rooms, or a donkey boiler above the bulkhead deck, and steam-pumps (workable from either source of steam) in two separate compartments, connected to the suctions.

8. **No sluice valve or cock** is to be fitted to the collision bulkhead.

9. **No sluice valves or cocks** are to be fitted to the engine-room or other watertight bulkheads unless they are arranged so as to be at all times accessible.

10. **When sluice valves** are fitted, they are to be so arranged as to be controlled above the load water-line, and the rods are to be boxed in to prevent injury.

11. **Sounding pipes** are to be fitted on each side of holds and ballast tanks, and a doubling plate is to be fitted under each.

12. **Air pipes** are to be fitted to each ballast tank as required.

13. All **cocks** and **valves** in connection with bilge and ballast suction pipes are to be fitted in places where they are at all times accessible.

14. All **bilge suction pipes** are to be fitted with strum boxes or strainers, so constructed that they can be cleared without breaking the joints of the suction pipes. The total area of the perforations in the strainers should not be less than double that of the cross section of the suction pipe.

15. The **filling pipes for deep tanks** which can be used for either cargo or ballast must be controlled by valves placed in an accessible position, and so arranged that when the tank is being used for cargo it will be impossible to fill it with water. This result is to be obtained by taking out a short bend or wedge piece and fitting blank flanges in its place, or in some other way to be submitted to and approved by the Committee.

16. The **pipes** for bilge or ballast suctions are to be fitted with flanged joints in convenient lengths, so that they may be easily disconnected for clearing. In the case of cast-iron suction pipes, which

are not also used as tank filling pipes, or which cannot be subjected to sea pressure, spigot and faucet joints made with india-rubber rings fitted over the spigots might be adopted, except in the case of bilge suction pipes passing through ballast tanks, which should be fitted with flanged joints.

17. The **suction pipes** to fore and after peaks, and to the tunnel well, should not be less than $2\frac{1}{4}$ inches inside diameter, except in vessels not exceeding 500 tons under deck, in which case they may be made 2 inches.

18. The **bilge injection** should not be less than two-thirds of the diameter of the sea inlet to the circulating pump.

19. The **inside diameter** of other **bilge suction pipes** should not be less than given in the following Table :—

Table CXXII.—Sizes of Bilge Suction Pipes by Lloyd's Rules.

Tonnage under upper deck.	Engine-room centre suction, separate donkey suction, and hold centre suctions.	Wing suctions in holds where no centre suctions are fitted, and wing suctions in engine-room.	Wing suctions in holds where centre suctions are also fitted.
	Inches.	Inches.	Inches.
In vessels not exceeding 500 tons, . . .	2	2	2
Above 500 tons but not exceeding 1000 tons, .	$2\frac{1}{4}$	2	2
Above 1000 tons but not exceeding 1500 tons, .	$2\frac{1}{2}$	$2\frac{1}{4}$	2
Above 1500 tons but not exceeding 2000 tons, .	3	$2\frac{3}{4}$	$2\frac{1}{4}$
Above 2000 tons but not exceeding 3000 tons, .	$3\frac{1}{2}$	3	$2\frac{1}{2}$
Above 3000 tons, . .	$3\frac{1}{2}$	$3\frac{1}{2}$	$2\frac{3}{4}$

In cases where more than one suction to any one compartment are connected to the pumps by a single pipe, this pipe should be not less than the size required for the centre suction.

LLOYD'S SURVEYS OF MACHINERY.

Lloyd's ordinary surveys of new machinery are thus described in the Rules :—

In steam vessels, the machinery and boilers are to be inspected throughout construction, the boilers tested by hydraulic pressure, and

the machinery tested under steam by the Society's Engineer-Surveyors, who will furnish a report to the Committee describing them in the manner shown in Form No. 4. If found satisfactory, the Committee will therefore grant a certificate, and insert in the Register Book the notification "LMC." in red (*i.e.* "Lloyd's Machinery Certificate"), indicating that the machinery and boilers are certified to be in good order and safe working condition.

Lloyd's Periodical Surveys (Section 19).

1. The machinery and boilers of all steam ships and the donkey boilers of sailing vessels are to be surveyed annually if practicable; and in addition are to be submitted to a Special Survey upon the occasions of the vessels undergoing the Special periodical Surveys Nos. 1, 2, and 3, prescribed in the Rules, unless the machinery and boilers have been specially surveyed within a period of twelve months.

2. At these Special Surveys, and on other occasions if deemed necessary by the Surveyors, the propeller, stern-bush, and the sea connections and their fastenings are to be examined while the vessel is in dry dock.

3. The stern shaft is to be examined annually, and drawn and examined at intervals of not more than two years.*

4. The cylinders, pistons, slide valves, crank and tunnel shafts, and pumps are to be examined, and if necessary the condenser is to be examined and tested.

5. The arrangements of cocks, pipes, bilge suctions, roses, &c., are to be examined.

6. The boilers and superheaters are to be examined internally and externally, and if deemed necessary by the Surveyors, both boilers and superheaters are to be drilled or tested by hydraulic pressure; the safe working pressure is to be determined by their actual condition.

7. The safety-valves are to be examined and set to the safe working pressure.

8. If satisfactory, these Surveys will be recorded in the Register Book thus:—"LMC7,08" *in red*, or "B&MS7,08" *in red*.

9. "LMC" (Lloyd's Machinery Certificate) denotes that the machinery and boilers are fitted in accordance with the Rules; and when followed by a date, indicates that they were found at that time to be in good condition. MS, with a date, denotes that the engines at that time were found upon inspection to be in good condition. BS, with a date, denotes that the boilers were found upon inspection at the time to be in good condition.

10. "B&MS" (Boilers and Machinery Surveyed), with a date, denotes that the boilers and machinery, though not fitted strictly in accordance with the rules, were found upon inspection at that time to be in good condition.

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* On the application of owners, the Committee will be prepared to give consideration to the circumstances of any special case.

11. In the event of either the machinery or boilers appearing to be impaired to such an extent as to render it desirable that either or both be specially surveyed within the periods prescribed above, a certificate for either machinery or boilers for a limited period will be granted according to the nature of the case.

Boilers.

12. The boilers of all steam ships and the donkey boilers of sailing vessels are to be specially surveyed when six years old, and subsequently they are to be specially surveyed annually.

13. At these surveys the boilers and superheaters are to be examined internally and externally, and if deemed necessary by the Surveyors, both boilers and superheaters are to be drilled or tested by hydraulic pressure; the safe working pressure is to be determined by their actual condition.

14. The safety-valves are to be examined and set to the safe working pressure.

15. If satisfactory, these surveys will be recorded in the Register Book thus:—"BS7,08" *in red* in the case of steam vessels, and "DBS7,08" *in red* in the case of sailing vessels.

16. "BS" (Boiler Surveyed) or "DBS" (Donkey Boiler Surveyed), with a date, denotes that the boilers were found upon inspection at that time to be in good condition.

17. In the event of the boilers appearing to be impaired to such an extent as to render it desirable that they be specially surveyed within the periods prescribed above, a certificate for a limited period will be granted according to the nature of the case.

For Lloyd's requirements as to steel castings, see under Cast Steel in section on "Materials."

LLOYD'S RULES RELATING TO SPARE GEAR.

Lloyd's requirements as to spare gear are as follows:—

The articles of spare gear mentioned in the following list will be required to be carried in all steam vessels classed in the Society's Register Book, viz.:—

2 connecting-rod, or piston-rod, top end, bolts and nuts.
2 ,, bottom end, bolts and nuts.
2 main bearing bolts.
1 set of coupling bolts.
1 set of feed and bilge pump valves.
1 set of piston springs (where common springs are used).
A quantity of assorted bolts and nuts.
Iron of various sizes.

In addition to the foregoing, the following articles are recommended

to be carried with a view to expedite repairs and lessen delay in distant ports, viz. :—

Crankshaft.
Propeller shaft.
Propeller, or a full set of blades.
Stern-bush, or lignum-vitæ lining for bush.
1 pair connecting-rod brasses.
1 pair crosshead brasses.
1 set of check valves.
6 cylinder cover bolts.
6 junk-ring bolts.
4 valve-chest cover bolts.
1 set of link brasses.
1 eccentric strap complete.
Air-pump rod.
Circulating-pump rod.
H.P. valve spindle.
L.P. ,,
2 dozen boiler tubes.
3 ,, condenser tubes.
1 cylinder escape valve and spring.
1 set of safety-valve springs.

MISCELLANEOUS BOARD OF TRADE RULES.

The following Board of Trade Rules concern the engineer, but cannot properly be placed under any of the preceding section headings:—

79. It is advisable that the donkey engine for pumping water through the condenser of the distilling apparatus be so fitted that it can also be made available in case of emergency for extinguishing fire in any part of the ship; a leather hose, with suitable bends and conductors, should be supplied for this purpose.

80. Each compartment of vessel should have an efficient hand-pump of sufficient size, workable from upper deck. When a double bottom is fitted extending the full length of a compartment and thereby preventing the deck-pump suction being placed near the shell plating at middle line of ship, a separate hand-pump or equivalent must be fitted to draw from waterway at each side of compartment. This regulation need not, however, be enforced in the case of fore and after peaks, chain lockers, or other small compartments near ends of vessel.

Suctions of hand-pumps should, whenever practicable, be placed at after end of compartments in which they are fitted.

When the required deck pumps are not situated on, or attached to, the upper deck, they should be of the closed-top type, and have their discharge pipes carried through the upper deck or through the ship's side, well above the deep-load line. If there are any cocks or valves in connection with the pumps which are not automatic in their action, suitable and efficient appliances should be provided, and kept permanently in position, whereby they may be manipulated from the upper deck.

The above rules as to the fitting of deck pumps with closed tops should be adhered to in new vessels, except in cases where the deck on which the pumps are situated has scuppers for conveying water that

may collect upon it to the outside of the vessel and is at such a height above the deep-load line as to preclude the probability of its being submerged in the event of the vessel getting holed. A rose or perforated box of sufficient size should be fitted to each suction pipe, and means should be provided for clearing it and the end of the pipe.

Deck pumps should be provided with suitable handles, and those of the smaller size should have handles long enough for at least two men to work them.

The Surveyor should be careful to see that the deck pumps in large vessels are not placed so high above the bottom of the compartments in which they are situated as to render them inefficient, and if he should at any time have doubts as to their efficiency he should test them.

It is very desirable that the steam winches (if any) be so fitted and arranged as to be available for working the hand deck pumps.

81. Rotary or fly-wheel pumps may be substituted for ordinary hand-pumps, provided that two such pumps are fitted, and that either is capable of pumping from any hold or machinery compartment in the vessel. They should be worked from a position well above the deep-load line, preferably from the upper deck; and unless they are placed in different watertight compartments, the pump buckets and valves should be capable of being removed from the position of working, so that they can be cleaned and repaired, if necessary, in the event of the compartment in which they are situated being flooded whilst the vessel is fully laden.

If the rotary hand-pumps are connected to the main bilge system, the various bilge suction valves or cocks should be provided with rods carried to a height well above the deep-load line, preferably to the upper deck. Index plates should be fitted at the upper ends to show what each valve or cock is for and how it is operated.

Hand-pumps may be dispensed with, if desired, in vessels fitted with two or more separate watertight boiler-rooms, and a separate watertight engine-room, provided that steam-pumps are fitted in at least two of the compartments named which are not adjacent, and are capable of pumping from any of the bilge suctions in the vessel, and provided also that the steam supply to the pumps is available from the boilers in either or any of the boiler compartments. The arrangement should be such that, in the event of a compartment being flooded, the bilge suction pipes to that compartment can be shut off, and that at least one of the bilge pumps may be available for pumping from any of the other hold or machinery compartments in the vessel.

82. Sounding pipes should be fitted from the upper deck for ascertaining the depth of water in each compartment.

The sounding pipes for the ballast tanks under the engine and boiler-room and under the tunnel floor may, however, be short, provided that they are fitted with screwed caps at the upper ends, or with cocks having the handles secured to the plugs.

83. Suction pipes connected with pumps worked by the main and donkey engines should be carried through the bulkheads into all the

compartments fore and aft of the engine-room, except the compartment in front of the collision bulkhead, so that each compartment can be pumped out separately by the engines. When a double bottom is fitted extending the full length of a compartment, with waterways on each side, a bilge suction should be fitted to each waterway. This requirement need not, however, be enforced in small compartments near the ends of the vessel. The pipes should be well secured where they pass through the bulkheads. In all new steamships, the cocks or valves which are fitted for the purpose of shutting off or controlling the flow of water through these pipes should, unless situated in the compartment occupied by the pump to which the pipes are connected, be provided with means by which they may be manipulated from a height well above the deep-load line, preferably the upper deck.

The arrangement of bilge suction pipes should be such that water cannot pass through them from one compartment to another. Valves in bilge distribution boxes should be of the non-return type, and if cocks are used instead of valves, suitable non-return valves should be fitted in the pipes to prevent water passing from one compartment to another in the event of the cocks being left open.

The free end of each hold suction pipe should be fitted with a suitable rose box or strum, and the tunnel-well suction pipe should be similarly fitted, unless it is provided with a mud-box of the form required for the pipes in connection with the engine-room and stoke-hold bilges.

Each main and donkey-engine suction pipe in connection with the engine-room and stoke-hold bilges should be provided with an efficient mud-box, or other similar appliance, placed above the platform, or in any other position in which it will be accessible at all times. The pipe leading from each mud-box to the bilge should, when practicable, be straight, and the cover of the mud-box should be secured in such a manner as to permit of it being expeditiously opened or closed.

The connection from the donkey engine to the bilge main should be by means of a switch cock or non-return valve.

An efficient bilge injection should be fitted to each main circulating pump.

A spare tiller (which has been properly fitted to the rudder-head), relieving tackle, &c., should, in all foreign-going and home-trade steamships, be kept near the after steering gear ready for immediate service. In large steamships where the use of hand gear is impracticable and such gear is not provided, the spare tiller should be attached to the rudder-head ready for immediate use, unless the working tiller is of special design and strength, in which case a spare tiller may not be required, but full particulars should be submitted for the Board's consideration. The steering gear, including chains, should be thoroughly overhauled at every survey, and taken to pieces and thoroughly examined at least once a year. The chains and blocks that are liable to interfere with or endanger the passengers or crew should be guarded by portable, but properly secured, guards.

With the view of relieving, as far as practicable, the rudders of

vessels from severe and sudden shocks, springs have in some cases been fitted to the quadrant, or to the rods or chains at each side of the vessel, and the Board think that such fittings, or other efficient means, should be adopted, more particularly in the case of new vessels.

The Surveyors should note that the steam and exhaust pipes of steering engines in all new passenger steamships should be at least of the same internal diameter respectively as the steam and exhaust connections on the cylinders. The arrangement should be such that water will not readily lodge either in the cylinders or in the steam and exhaust pipes. Right-angled bends in the pipes should be avoided as much as possible, and the pipes should be used exclusively for the steering engines. When this is not the case, full particulars and sketches should be submitted to the Board for consideration.

Attention is also directed to a description of steam-steering gear in which a part of the shaft, by which the helmsman actuates the controlling valve, passes through another shaft that is liable to be thrown out of line by the reaction of the spur gearing, and, consequently, is liable to jam the inside shaft to such an extent as to deprive the helmsman of the control of the steering gear. All steam steering gears should be carefully examined, and if any be found constructed in the manner described above, their use should be discouraged, and in any case they should not be approved, unless they have been tested from midship to hard over in both directions, and found satisfactory when the vessel is running at full speed.

It is very desirable that the man at the helm should be so placed that he has a clear lookout ahead, more especially in steamers that frequent crowded harbours or rivers; and, in case of steam or electric launches, the Surveyor must assure himself that view cannot be obstructed by passengers.

77. In passing a helm indicator, the Surveyor should ascertain by actual trial that whenever the pointer moves to the word "port" on the dial or plate, wherever that word may be, it shows that the helm is ported, and whenever the pointer points to the word "starboard," wherever that word may be, it means that the helm is starboarded.

In the foregoing directions it is assumed that "port helm" means that the helm is so moved as to turn the ship's head to the right, and "starboard helm" means that the helm is so moved as to turn the ship's head to the left.

78. Passenger steamers going to sea should be provided with a hose adapted for the purpose of extinguishing fire in any part of the ship, and capable of being connected with the engines of the ship, or with the donkey engine, if it can be worked from the main boiler. The Surveyor must take care that it answers the required purpose.

The fire hose should be connected and stretched, to judge of its length, and thoroughly examined at every survey, and at least once a year (and at any other time that the Surveyor thinks it necessary) it should be tested with the conductor in its place by pumping water through it by the main or donkey engines at full speed. A proper conductor and metal bend or goose neck form part of its equipment,

and should be provided. Generally, leather hoses are the most durable, and should be supplied when a declaration for 12 months is required.

191. **Cast-iron stand-pipes, cocks, &c.,** intended for passage through them of hot brine, should not be passed. Surveyors should also discourage the use of cast iron for chocks and saddles for boilers. Particular attention should be paid to the chocking of boilers, more especially when they are fired athwartships.

200. In the case of steamers performing ocean voyages, and coming in for survey, no question as to gear need be raised if the following spare gear and stores are supplied, or their equivalent, particulars of which should be submitted to the Board for consideration. The heavier portions of this gear should have been fitted and tried in their places, and should be kept on board where access can at all times be had to them:—

1 pair of connecting-rod brasses.
1 air-pump bucket and rod, with guide.
1 circulating-pump bucket and rod.
1 air-pump head valve, seat, and guard.
1 set of india-rubber valves for air-pumps or ⅛ set of metallic ones.
1 circulating pump head valve, seat, and guard.
1 set of india-rubber valves for circulating pumps, or ⅛ set of metallic ones.
2 main bearing bolts and nuts.
2 connecting-rod bolts and nuts.
2 piston-rod bolts and nuts.
8 screw shaft coupling bolts and nuts.
1 set of piston springs suitable for the pistons.
1 set metal feed-pump valves and seats.
3 sets if of india-rubber, or 1 set if of metal, of bilge-pump valves and seats.
1 spring (at least) for each size of escape valve.
1 hydrometer.
Boiler tubes, 3 for each boiler.
100 iron assorted bolts, nuts, and washers, screwed, but need not be turned.
12 brass bolts and nuts, assorted, turned, and fitted.
50 iron ,, ,, ,,
50 condenser tubes.
100 sets of packing for condenser tube ends, or an equivalent.
At least one spare spring of each size for escape valves.
1 set of water-gauge glasses.
$\frac{1}{10}$th of the total number of fire-bars necessary.
3 plates of iron, assorted.
6 bars of iron, assorted.
1 complete set of stocks, dies, and taps, suitable for the engines.
100 iron assorted bolts, nuts, and washers, screwed, but not necessarily turned.

12 brass assorted bolts, nuts, turned and fitted.
50 iron " " "
1 smith's anvil.
1 fitter's vice.
Ratchet braces, and suitable drills.
1 copper or metal hammer.
Suitable blocks and tackling for lifting weights.
1 dozen files, assorted, and handles for the same.
1 set of drifts or expanders for boiler tubes.
1 safety-valve spring (if so fitted) for every four valves if of the same size; but more than 6 spare springs of the same size need not be provided, whatever be the number of the valves; if there are not four valves, then at least one spare spring must be carried.
1 screw jack.
And a set of engineer's tools suitable for the service, including hammers and chisels for vice and forge; solder and soldering iron; sheets of tin and copper; spelter; muriatic acid, or other equivalent, &c.

147. The following list of tools and materials should be provided for distilling apparatus:—

1 set of stoking tools.
1 scaling tool.
1 spanner for boiler doors.
1 set of fire-bars, suitable for boiler.
1 14-inch flat bastard file.
1 14-inch half-round file.
1 10-inch round file.
3 file handles.
2 hand cold chisels.
1 chipping hammer.
1 pair of efficient gas tongs.
1 soldering iron.
10 lbs. of solder.
2 lbs. of rosin.
6 gauge glasses.
24 india-rubber gauge-glass washers.
30 bolts and nuts, assorted.
1 slide rod for donkey pump.
5 lbs. spun yarn.
10 lbs. cotton waste.
1 deal box, with lock, complete.
2 gallons machinery oil.
Animal charcoal sufficient to charge the filter at least twice.
1 can for machinery oil.
1 oil feeder.
1 small bench vice.
1 ratchet brace.
4 drills, assorted.
1 set dies and taps suitable for the bolts.
2 glass salinometers.
1 hydrometer and pot.
1 shifting spanner.
1 lamp for engineer.
And other articles that the particular distiller and boiler supplied may, in the Surveyor's judgment, require.

197. The boiler for supplying steam to the distillers of an emigrant ship should be at least equal in strength to the boilers of passenger steamships, and should have fittings in accordance with these regulations. The steam for working the apparatus is not to be taken from the main boilers. No exhaust steam may be permitted to go into the

condenser, if appliances for the introduction of lubricants are fitted to the steam pipes or steam cylinder of the pumping engine. The boiler of the apparatus must not be filled or fed with water from the surface condensers of the main engines, and must not be fitted with cocks, &c., for the introduction of tallow or oil. The presence of zinc in such boilers is also objectionable, and the Surveyors should refuse to pass boilers so fitted for use on board vessels clearing as emigrant ships.

No distilling apparatus should be passed unless fitted with a filter of suitable size charged with animal charcoal.

When the water is pumped into the condenser, the latter should be fitted with an efficient escape-valve which cannot readily be tampered with; and, if the condensing portions of the apparatus or the cooler and filter are unfit to bear the pressure on the boiler, an efficient safety-valve which cannot readily be overloaded, should be fitted between the steam pipe and the apparatus.

The Surveyor should satisfy himself as to the capability of the man who is to have charge of the apparatus.

As the Surveyor will be held wholly responsible for the efficiency of the distillers, it rests with him to decide whether or not the apparatus should, in the case of emigrant vessels furnished with passenger certificates, be taken to pieces for examination prior to every voyage, but the distilling apparatus of such vessels must be taken to pieces for examination at least once every twelve months, or more often if the Surveyor thinks it necessary. The tubes or coils should be tested to at least twice the load on the safety-valve on the apparatus, or, in cases where no safety-valve is fitted, to twice the highest working pressure of the boiler from which the apparatus can be worked, and the machinery and boiler should be thoroughly examined. The charcoal should be removed from the filter, cleansed, or renewed, at least once every six months.

After the distilling apparatus is put together again, it should be tested as to the quantity and quality of the water made, and this should also be done before the commencement of every voyage. The water should be cool, pure, and fit to drink immediately it is drawn off from the filter.

The presence of zinc in such boilers is also objectionable, and Surveyors should refuse to pass boilers so fitted for use on board vessels sailing under the Merchant Shipping Act of 1894, Part III.

When the water is pumped into the condenser there should be an efficient escape-valve on the condenser, which cannot be readily tampered with, and if the condensing portion of the apparatus or the cooler and filter be unfit to bear the pressure on the boiler, an efficient safety-valve that cannot be readily overloaded should be fitted between the steam pipe and the apparatus.

146. It is advisable that the donkey engine for pumping water through the condenser be so fitted that it can be made available in case of emergency for extinguishing fire in any part of the ship; a leather hose, with suitable bends and conductors, should be applied for this purpose.

CHAINS AND ROPES

The following Tables give the Admiralty requirements as to chains:—

Table CXXIII. Admiralty Tests, &c., of Stud-link Chain Cable.

Dia. of cable in inches.	Breaking strength in tons.	Proof load in tons.	Weight of 100 fathoms in cwts.	Dia. of cable in inches.	Breaking strength in tons.	Proof load in tons.	Weight of 100 fathoms in cwts.
7/16	4·90	3·5	9·25	1½	56·70	40·5	108·0
½	6·30	4·5	12·0	1⅝	66·50	47·5	126·75
9/16	7·70	5·5	15·25	1¾	77·17	55·125	147·0
⅝	9·80	7·0	18·75	1⅞	88·55	63·25	168·75
11/16	11·90	8·5	22·75	2	100·80	72·0	192·0
¾	14·17	10·125	27·0	2⅛	113·75	81·25	216·75
⅞	19·25	13·75	36·75	2¼	127·57	91·125	243·0
1	25·20	18·0	48·0	2⅜	142·10	101·5	270·75
1⅛	31·85	22·75	60·75	2½	157·50	112·5	300·0
1¼	39·37	28·125	75·0	2¾	181·02	129·3	363·0
1⅜	47·60	34·0	90·75	3	204·12	145·8	432·0

Up to and including 2½ inches diameter, the above proof loads are equal to 630 lbs. per circular ⅛ inch of section of one side of link, and are the same as those required by the Chain Cables and Anchors Acts; but for 2¾ inches diameter the load is reduced to 598·5 lbs. per circular ⅛ inch, and for 3 inches diameter to 576 lbs. The formula for proof loads of chains up to 2½ inches diameter may also be written,—

$$\text{Proof load in tons} = 18 \times (\text{diameter in inches})^2.$$

The breaking strengths are placed at 40 per cent. above the proof loads.

Table CXXIIIa.—Admiralty Tests, &c., of Short-link Chain.

Dia. of chain in inches.	Breaking strength in tons.	Proof load in tons.	Weight per fathom in pounds.	Dia. of chain in inches.	Breaking strength in tons.	Proof load in tons.	Weight per fathom in pounds.
¼	1·87	·75	3·0	15/16	26·37	10·55	49·0
5/16	2·93	1·17	5·5	1	30·00	12·00	56·0
⅜	4·22	1·69	8·0	1 1/16	33·87	13·54	63·0
7/16	5·74	2·30	10·5	1⅛	37·97	15·18	71·0
½	7·50	3·00	14·0	1 3/16	42·30	16·92	79·0
9/16	9·49	3·80	18·0	1¼	46·87	18·75	87·0
⅝	11·72	4·69	22·0	1 5/16	51·68	20·67	96·0
11/16	14·18	5·67	27·0	1⅜	56·72	22·68	106·0
¾	16·87	6·75	32·0	1 7/16	62·00	24·80	116·0
13/16	19·80	7·92	37·0	1½	67·50	27·00	127·0
⅞	22·97	9·19	43·0	...	...	...	...

The proof loads for short-link chains are ⅔rds those for stud-link chains.

The rule may be written,—**Proof load in tons = 12 × (diameter in inches)²**. The breaking strengths of short-link chains are placed at 2½ times the proof loads.

Lloyd's requirements as to chain cables, &c., are given in Table CXXIV.; the proof loads and breaking strengths are those required by the Act of Parliament; the breaking strengths for chains above 1½ inches diameter are the same as required by the Admiralty, but for chains of 1½ inches diameter and under, Lloyd's require a slightly higher breaking strength.

Table CXXIV.—Lloyd's Tests of Stud-link Chain Cables.

Diameter of chain in inches.	Proof load (Statutory) in tons.	Breaking strength (Statutory) in tons.	Diameter of chain in inches.	Proof load (Statutory) in tons.	Breaking strength (Statutory) in tons.
11/16	8·5	12·75	1⅝	47·5	66·5
¾	10·125	15·125	1 13/16	51·25	71·75
13/16	11·875	17·8	1¾	55·125	77·125
⅞	13·75	20·625	1 15/16	59·125	82·75
15/16	15·8	23·7	1⅞	63·25	88·5
1	18·0	27·0	1 15/16	67·5	94·5
1 1/16	20·3	30·4	2	72·0	100·8
1⅛	22·75	34·125	2 1/16	76·5	107·1
1 3/16	25·375	38·0	2⅛	81·25	113·75
1¼	28·125	42·125	2 3/16	86·125	120·5
1 5/16	31·0	46·5	2¼	91·125	127·5
1⅜	34·0	51·0	2 5/16	96·25	134·75
1 7/16	37·125	55·625	2⅜	101·5	142·1
1½	40·5	58·7	2 7/16	106·9	149·625
1 9/16	43·9	61·4	2½	112·5	157·5

Note.—Unstudded close-link chains will be admitted as cables if proved to two-thirds the load required for stud-link chains, and if the breaking strength is not less than twice such proof load.

In some recent tests of the chain cables of large steamers a 2⅝-inch cable gave an average ultimate strength of 212 tons, and a 2 9/16-inch cable gave 223 tons as the lowest and 229 tons as the highest of seven tests.

The safe working load on chains should not be taken higher than half the proof load; if this proportion be adopted the formula becomes:—

Working load in tons = { 9 × (diameter in inches)² for stud-link. / 6 × (diameter in inches)² for close-link.

For ordinary crane chains and slings 4 × (diameter in inches)² is high enough.

Table CXXV.—Admiralty Flexible Steel Wire Ropes.

Size of rope (circumference) in inches.	Number of wires in each strand.	Weight per fathom in lbs.	Minimum breaking st'gth in tons.	Torsion test; No. of twists each wire must stand.	Ductility tests, &c.
8	30	53	148	9	Each rope to consist of 6 strands. Wires to be of best crucible steel galvanized with pure zinc.
7	30	41	113	11	
6½	30	35	98	14	The rope is to be laid up evenly and uniformly as regards size and angle, and is to contain a proper sized hemp core.
6	30	31	84	15	
5½	24	28	71	16	
5	24	23	59	17	A latitude not exceeding 5 per cent. over or under the prescribed weights will be allowed.
4½	12	14	39	15	
4	12	12	31	17	Elongation will be assumed to commence when one-sixth of the breaking-load has been applied.
3½	12	9	24	18	
3	12	7	17	22	**Test A** (torsion).—Each wire to stand being twisted through the number of revolutions stated in column 5, in a length of 8 inches.
2¾	12	5½	14½	25	
2½	12	4½	11¾	26	
2¼	12	3¾	9	28	**Test B** (bending).—Each wire to stand coiling around itself eight turns and back again.
2	12	2¾	7	33	
1¾	12	2	5½	36	If, in being tested, one wire in seven fail, but the other six give fair and uniform results, the average being up to the standard, the rope will be considered satisfactory in that respect.
1½	12	1¾	4	41	
1¼	12	1¼	2⅞	47	
1	12	¾	1¾	60	

Proportions of Links of Chains.—The standard proportions of the links of chains, in terms of the diameter of the bar from which they are made, are as follows :—

	Overall length.	Overall breadth.
Stud-link	6 diameters	3·6 diameters
Close-link	5 ,,	3·5 ,,

The stud has usually a diameter at the centre of ·6 × diameter of chain, and at the ends 1 × diameter of chain.

Weight of Chains.—The weight of stud-link chain cables is given very nearly by the rule,—

$$W = 55d^2,$$

where W = weight per fathom in pounds, and d = diameter of bar from which chain is made.

For close-link chain the rule becomes,—

$$W = 58d^2.$$

The above rules give weights about midway between those required by the Admiralty and those required by Lloyd's. The weight is of course much affected by the length of link used.

Steel Wire Ropes.

The Admiralty requirements with regard to flexible steel wire ropes are given in Table CXXV.

Steel wire ropes for standing rigging are required to be made of fewer wires of larger diameter, are rather heavier, and must be of rather greater ultimate strength.

The following Table shows the breaking strengths that steel wire hawsers, &c., must show in order to be accepted by Lloyd's :—

Table CXXVI.—Breaking Strengths of Steel Wire Hawsers, &c., by Lloyd's Rules.

Size (circumference) in inches.	Breaking strength in tons.	Size (circumference) in inches.	Breaking strength in tons.	Size (circumference) in inches.	Breaking strength in tons.
2	7	3¾	29	5½	71
2¼	9½	4	33	5¾	78
2½	12¼	4¼	35	6	85
2¾	15½	4½	39	6½	100
3	18	4¾	47	7	113
3¼	22	5	59	7½	128
3½	26	5¼	65	8	149

Table CXXVIa.—Special Flexible Steel Wire Rope.

Size (circumference) in inches.	Lloyd's Breaking Test in tons.	Size (circumference) in inches.	Lloyd's Breaking Test in tons.
1¾	8·9	3¾	41·0
2	11·7	4	52·5
2¼	14·6	4¼	59·0
2½	18·2	4½	65.5
2¾	22·0	5	73·0
3	26·2	5½	88·0
3¼	30·7	6¼	114
3½	35·5		

A short length of each of the wires composing the hawser, &c., will also be required, after being galvanised, to show a tensile strength equivalent to that given in the above table, and the aggregate strength of the wires must not be less than 10 per cent. in excess of that strength.

Each wire must also be capable of being twisted around itself not less than eight times, and of being untwisted and straightened again without breaking.

The strength of steel wire ropes, relatively to their girth, depends not only on the quality of the wire used, but also on the amount of hemp core used; usually there is a central hemp core, and sometimes each strand has also a similar core, but sometimes there are no hemp cores at all. There would be no particular difficulty in obtaining ropes to stand twice the tests given in the above Table, but such ropes would probably be wanting in flexibility, and would require drums of very large diameter if they were to work satisfactorily and to last any time.

Hemp Ropes.

Hemp is laid up right-handed in yarns; and yarns are laid up left-handed into strands.

A hawser is composed of three strands laid up right-handed.

A cable is composed of three hawsers laid up left-handed.

Shroud-laid rope has a core surrounded by four strands.

The strength of hemp ropes depends on the quality of the hemp used, on the type or make of the rope, and on its condition (*i.e.* wet or dry, tarred or untarred).

The twist diminishes the strength, but increases the solidity and durability, and the strength therefore depends to some extent on the twist.

When a rope is wet or tarred its strength is reduced by about one-fourth.

The working strength is commonly taken as ⅛th of the breaking strength.

Table CXXVII.—Bullivant's Steel Wire Ropes (Galvanised)

For Ships' Hawsers, Running Rigging, Cargo Falls, etc.

TABLE SHOWING WEIGHTS AND BREAKING STRESSES.

Flexible Steel Wire Rope, 6 Strands, each 12 Wires.					Extra Flexible Steel Wire Rope, 6 Strands, each 24 Wires.		Special Extra Flexible Steel Wire Rope.		
							6 Strands, each 37 Wires.		Special Make.
Size Circumference.	Size Diameter.	Approx. Weight per Fathom.	Approx. Breaking Stress (Actual).	Diamr. of Barrel or Sheave round which it may be worked at a slow speed.	Approx. Weight per Fathom	Approx. Breaking Stress (Actual).	Approx. Weight per Fathom.	Approx. Breaking Stress (Actual).	Approx. Breaking Stress (Actual).
Ins.	Ins.	Lbs.	Tons.	Ins.	Lbs.	Tons.	Lbs	Tons.	Tons.
¾	¼	·420	1·1	4·5	·488	1·5	...	...	...
1	5/16	·720	2·0	6	·960	2·8	1·08	3·0	...
1¼	...	1·08	3·1	7·5	1·44	4·3	1·50	4·0	...
1½	...	1·50	4·4	9	1·98	6·0	2·16	6·6	...
1¾	9/16	2·10	5·9	10·5	2·82	8·5	3·00	8·5	...
2	⅝	2·70	8·0	12	3·60	10·9	3·96	11·3	...
2¼	...	3·60	10·5	13·5	4·68	14·2	5·04	14·5	...
2½	13/16	4·20	12·3	15	5·70	17·2	6·12	18·2	...
2¾	⅞	5·10	14·8	16·5	6·78	20·5	7·38	22·2	...
3	15/16	6·00	17·6	18	8·04	25·0	8·70	26·6	...
3¼	...	7·30	21·1	19·5	9·54	28·9	10·32	29·0	...
3½	1⅛	8·40	24·4	21	11·28	34·2	11·88	34 0	...
3¾	...	9·60	27·8	22·5	12·78	38·8	13·80	39·5	...
4	...	10·98	32·3	24	14·34	43·6	15·78	45·3	...
4¼	...	12·00	35·4	25·5	16·50	50·1	17·70	51·6	...
4½	...	14·04	41·3	27	18·72	57·0	19·98	58·2	...
4¾	1½	...	...	...	20·64	62·8	22·14	65·3	...
5	...	...	...	...	22·68	68·9	24·54	72·7	...
5¼	...	...	...	...	24·78	75·3	27·18	80·6	...
5½	1⅜	...	...	...	27·00	82·0	29·58	86·7	...
5¾	...	...	...	...	...	...	32·28	93·1	...
6	1⅞	...	...	...	...	...	35·40	104·3	...
6½	...	...	...	...	...	...	40·92	118·5	...
7	...	...	...	...	...	...	47·94	136·2	...
7½	...	...	...	...	...	...	55·00	178·5	...
8	...	...	...	...	...	...	63·00	198·0	202
9	...	...	...	...	...	...	79·00	250·0	257
10	...	...	...	...	...	...	98·00	305·0	318
11	...	...	...	...	...	...	120·00	...	381
12	...	...	...	...	...	...	142·00	...	455

In these Flexible Rope Tables the wire is calculated as taking a breaking stress of 90 tons to the square inch.

Messrs BULLIVANT & CO., LTD., are prepared to make steel ropes to take a *breaking stress of* 1200 *tons on the single part*, and they will furnish any information connected with larger size cables than those mentioned in the above Table.

Table CXXVIII.—Bullivant's Mild Plough Steel Wire Crane Ropes (Black).

Flexible Steel Wire Rope, 6 Strands, each 12 Wires.				Extra Flexible Steel Wire Rope, 6 Strands, each 24 Wires.		Special Extra Flexible Steel Wire Rope, 6 Strands, each 37 Wires.	
Size Circumference.	Approx. Diam.	Approx. Weight per Fathom.	Approx. Breaking Stress.	Approx. Weight per Fathom.	Approx. Breaking Stress.	Approx. Weight per Fathom.	Approx. Breaking Stress.
Ins.	Ins.	Lbs.	Tons.	Lbs.	Tons.	Lbs.	Tons.
1	$\frac{5}{16}$	·8	1·9	·9	3·0	1·1	3·2
$1\frac{1}{4}$	...	1·1	2·9	1·5	5·0	1·6	4·8
$1\frac{1}{2}$	...	1·6	4·1	2·0	7·0	2·4	7·1
$1\frac{3}{4}$	$\frac{9}{16}$	2·1	5·6	2·7	9·3	3·2	9·9
2	$\frac{5}{8}$	2·9	7·6	3·7	12·7	4·3	13·2
$2\frac{1}{4}$	...	3·6	9·5	4·5	15·8	5·2	17·0
$2\frac{1}{2}$	$\frac{13}{16}$	4·4	11·7	5·5	19·2	6·4	20·1
$2\frac{3}{4}$	$\frac{7}{8}$	5·3	14·0	6·9	23·9	7·9	24·7
3	$1\frac{1}{16}$	6·2	16·6	8·0	28·1	9·2	29·8
$3\frac{1}{4}$	...	7·5	20·0	9·7	33·8	11·0	33·8
$3\frac{1}{2}$	$1\frac{1}{8}$	8·6	23·0	11·1	38·7	12·9	39·7
$3\frac{3}{4}$	...	9·8	26·3	12·5	43·9	14·8	46·0
4	...	11·1	29·8	14·5	50·9	16·5	52·9
$4\frac{1}{4}$	...	12·5	33·6	16·2	56·9	18·7	60·1
$4\frac{1}{2}$	...	14·2	38·2	18·0	63·2	21·0	65·9
$4\frac{3}{4}$	$1\frac{1}{2}$	15·8	42·4	20·4	71·6	23·5	74·1
5	...	17·4	46·8	22·4	78·7	25·7	82·6

Note.—The diameter of barrel or sheave for flexible ropes should not be less than six times the circumference of the rope, but where extra flexible and special extra flexible ropes are used the diameter may be somewhat reduced.

The Admiralty requirements as to various sizes of hawser-laid cordage in common use are given in Table CXXIX.; the 11-inch rope is specified to be of tarred Petersburg hemp, but all the smaller sizes are to be of tarred Riga hemp; all are to be three-strand.

Table CXXIX.—Admiralty Tarred Hemp Cordage.

Size of rope (circumference) in inches.	Size of yarn.	Number of threads in the rope.	Standard breaking strength.		
			Tons.	cwt.	qrs.
½	40	6	0	3	0
¾	,,	12	0	6	0
1	,,	15	0	8	0
1½	,,	33	0	15	0
1¾	,,	42	1	0	0
2	,,	54	1	7	0
2½	,,	84	2	0	0
3	,,	120	3	0	0
3½	30	123	3	18	0
4	,,	159	5	0	0
4½	,,	201	6	9	0
5	,,	249	7	18	0
11	25	1008	36	10	0

All sizes are specified to be formed at an angle of 27°, hardened at 37°, and finished at 42°.

If the smallest size be excluded, the above Table gives breaking strengths which approximate very closely to those given by the formula,—

$$\text{Breaking strength in cwts.} = (6{\cdot}2 \times \text{girth in inches}^2) + 2.$$

The weight in pounds per fathom of hawser-laid hemp ropes is given approximately by the rule:—

Weight in lbs. per fathom = $\cdot 17 \times (\text{girth in inches})^2$ for dry ropes.

,, ,, = $\cdot 21 \times (\text{girth in inches})^2$ for wet or tarred ropes.

STRENGTH, &c., OF MATERIALS.

Generally, where reference is made in this section to the extension per cent. of a test piece, it is to be understood that the piece is 8 diameters long between the marks, if of circular section, or 12·75 thicknesses long by 2 in width, if of rectangular section.

Cast Iron.

The strength that an iron casting may be expected to possess depends on the quality of the iron, on the number of times it has been melted, on the design or form of the casting, and on the skill and care exercised in the foundry to ensure soundness and freedom from contraction stresses. The quality of the iron depends on its chemical constitution, and on the method of manufacture. The effects of variations in chemical composition are indicated in the following table:—

Table CXXX.—Compositions and Qualities of Cast Iron.

Quality.	Composition.		
	Combined carbon per cent.	Graphitic carbon per cent.	Silicon per cent.
Very soft, . . .	·15	3·1	2·5
Very hard, . . .	...	...	under ·8
Great general strength, .	·5	2·8	1·42
Great tensile strength, .	...	...	1·8
Great crushing strength,	over 1·0	under 2·6	about ·8

Pig irons are as a rule divided into grades, in number sometimes seven but often only three, each of which is known by a number. No. 1, grey iron, contains the most free or graphitic carbon, and, when broken, exhibits a very coarsely granular fracture with dark grey scales of considerable size; when melted, "it runs very thin," or is of extreme fluidity, and is therefore used mostly for fine ornamental castings, and for mixing with other numbers when increased fluidity is required.

The highest number, whether there be seven or three grades, is a *white iron*, and contains the carbon in a combined state with little or none free. The middle number or numbers have the carbon partly free and partly combined, and called *mottled iron* from the appearance of fracture.

When in seven grades, No. 2 pig is neither so soft nor so fluid as No. 1, but is not sufficiently close grained for general use.

No. 3 pig is that usually employed for marine engine castings; by adding No. 1 a mixture suitable for smaller complicated castings is obtained, and the addition of No. 4 gives a harder and closer-grained metal.

No. 4 is almost a forge iron, not much used in the foundry except for such mixing purposes; it still shows a mottled grey fracture, but the grain is finer and more crystalline, and there is an absence of the graphitic scales so marked in Nos. 1, 2, and 3.

Nos. 5 and 6 are not used in the foundry at all, but are made for conversion into wrought iron, &c.

No. 7 pig shows a silvery white and crystalline fracture, contains practically no free carbon, and is extremely hard. Like Nos. 5 and 6, it is a "forge iron," and is sometimes called "white forge," while Nos. 5 and 6 are called "grey forge."

Iron from different districts, and made from different classes of ore, of course varies considerably in composition and quality, but quality is also considerably affected by the method of reduction,—cold-blast iron being usually stronger, tougher, and closer grained than hot-blast, and therefore often used for mixing with other irons where exceptional strength and toughness is required.

Iron Mixtures.—As all cast irons are improved by mixing as well as re-melting, no important casting should be made entirely of new pig, and if maximum strength is required, the whole of the material should be re-melted. Chemical analysis should be made and the same composition aimed at by judicious mixing to get uniformly good results.

For cylinders a strong, tough, and close-grained metal is required, which may be obtained by using equal proportions of picked scrap, best Scotch No. 3 pig, and good cold blast. If the cylinder is to have liners and false faces, most of the hardening elements may be reduced.

Cylinder liners and false faces require a good amount of strength and great hardness, and may be made of a similar mixture hardened up with some No. 4 pig.

Since cold-blast irons are scarce, and always dear, hematite with good Scotch or other equally suitable irons may be used for such purposes. In fact, hematite is often used with No. 1 and No. 3 irons of quite ordinary brands for cylinders, &c., of cargo-boat engines when weight is not of prime importance.

Castings of simple form, such as propeller blades and bosses, may be increased in hardness, strength, and closeness of grain by the addition of steel boiler plate scrap to the extent of 10 per cent. or even higher. Hematite is used also for the same purpose.

The contraction of iron castings in cooling varies considerably with the form and proportions of the casting, but is, on an average, $\frac{1}{10}$ to $\frac{1}{8}$ inch per foot.

The cooling of large and intricate castings should be as gradual as possible, as the internal stresses are then somewhat relieved, and the risk of cracks diminished. The slower the cooling is the larger will the crystals be, and *vice versa*, but slow cooling becomes a kind of annealing.

The weight of cast iron also varies very considerably,—the difference between the heaviest and lightest kinds being nearly 40 lbs. per cubic foot; but a fair average value, and one easily remembered, is about 450 lbs. per cubic foot, or ·26 lbs. per cubic inch. A plate 1 foot square × 1 inch thick will then weigh 37·5 lbs. Ordinary marine castings are probably rather above than below this weight.

Strength of Cast iron.—Cast iron suitable for ordinary marine castings should not have a lower ultimate tensile strength than 17,000 lbs. per square inch, and when weight is of importance, and scantlings

are cut down, a strength of 22,400 lbs. per square inch should be aimed at, as now with judicious blending of brands, additions of steel scrap, and using coke free from sulphur, even 36,000 lbs. can be attained.

Testing, both chemical and mechanical, should be constantly resorted to if there is to be any check on the foundry, or if any accurate knowledge as to the material being turned out is required.

The ultimate strength of cast iron in compression is about 90,000 lbs. per square inch; in ordinary construction it may carry three times as much as in tension.

The Admiralty requirements as to cast iron are as follows:—

Test pieces to be taken from such castings as the inspecting officer may consider necessary. The minimum tensile strength to be 9 tons (20,160 lbs.) per square inch, taken on a length of not less than 2 inches.

The transverse breaking load for a bar 1 inch square, loaded at the middle between supports 1 foot apart, is to be not less than 2000 lbs.

Wrought Iron.

The quality of wrought iron,—provided it is free from such harmful ingredients as sulphur and phosphorus,—depends largely on the amount of work that has been done on it at the mill,—*i.e.* on the extent to which it has been rolled down and the so-called fibre developed.

The appearance of the fracture depends a good deal on the manner in which it is broken; if good bar iron is nicked at one side, and slowly broken or bent back, it should show a clear white silvery, and almost entirely fibrous, fracture, whereas if nicked on two sides, or broken with a smart severe blow, the fracture will show more white crystalline grain, with little or no fibre.

A specimen from a good forging will show a clear and silvery grain, but larger than that of the bar iron, with about 20 to 30 per cent. of fibrous patches.

An inferior iron usually shows a coarse crystalline structure, or if fibre is present, it is dull and earthy looking.

Coarse crystals, or large shining plates, generally indicate a "cold-short" iron, and "red-shortness" is indicated by an earthy, dull, or dark fracture.

Merchant bar is the commonest quality formerly used for ladders, gratings, fire-bars, bearer bars, &c., but steel is now generally cheaper and much better for these purposes.

Best bar is the next quality; its tensile strength is about 24 tons (or, say, 54,000 lbs.) per square inch, and it may be used for all ordinary smithing purposes.

Best best bar is a higher quality again, and has an ultimate tenacity of 26 to 27 tons (or 58,000 to 60,000 lbs.) per square inch, with an elongation of about 25 per cent. in 8 inches, and a contraction of area of about 50 per cent.; the fibre is uniform and silky in appearance, and the bar may be bent double, cold, without fracture.

It is still largely used for the screwed stays of boilers,* as it stands better than steel the strains coming on them; the Board of Trade permit the use with a working stress of 9000 lbs. per square inch, providing the iron is tested as is steel. Bar steel has largely taken the place of iron, but for smithing purposes or for chain-making where welding is resorted to, best Yorkshire and Staffordshire bars, and bars made from boiler plate scrap, remain still in quite good demand.

Weight of wrought iron.—Wrought iron varies slightly in density according to the method of manufacture, the form into which it is put, and the amount of impurity it contains. A fair average value is 485 lbs. per cubic foot, or ·28 lb. per cubic inch; bars and Yorkshire plates give about this figure, but Staffordshire plates are rather lighter (about 480 lbs.), and large forgings lighter again (about 477 lbs.),—owing probably to the presence of cinder in a minutely divided condition. A square foot of plate, one inch thick, usually weighs 40 lbs.

Cast Steel.

Steel castings for the pistons, covers, framing, &c., of marine engines should have an ultimate tensile strength of 29 to 31 tons (65,000 to 69,000 lbs.) per square inch, with an extension of about 12 to 15 per cent in 8 inches. For castings that are at all intricate, or thin in parts, it is necessary to use a steel with more carbon and a strength up to 36 or 38 tons ultimate, as the milder and tougher steels are not sufficiently fluid when melted, but then the extension obtainable will often not exceed 8 per cent. The ultimate tensile in tons per square inch added to extension per cent. should be not less than 45. British Engineering Standard is 50.

The contraction of cast steel in cooling is more variable than that of cast iron, but is on an average about $\frac{3}{16}$ inch per foot, or the same as that of brass. Special care should therefore be taken in designing large pieces to be cast in this metal, and forms that will interfere with the contraction, or cause the casting to "draw" in cooling should be avoided; when possible an O pen or H section should be preferred to a close or box section for framings, as the former is more likely to give a sound casting,—especially if plenty of small fillets are placed in the angles (say $\frac{1}{8}$ inch thick and 6 inches pitch—for large framings—dying away at 3 inches out from the angle in both directions) and a good radius is used.

Annealing, for the purpose of relieving internal stresses set up, is absolutely necessary for large or intricate castings, and desirable for all.

Soft steels of this class will contain up to ·3 per cent. of carbon; when the proportion reaches about 1 per cent. it becomes possible to harden or temper the steel.

Good hard cast steel, with an ultimate strength of 50 or 55 tons, will probably contain 1·3 or 1·4 per cent. of carbon.

* B. of T. tests for iron bars, 21·5 tons tensile with 25% elongation in 8 diameters.

To weld properly, steel must not contain more than ·25 per cent. of carbon. Silicon steel of quite high tensile strength will weld readily.

Weight of steel castings.—Soft steels, of 28 to 35 tons ultimate strength, weigh about 490 lbs. per cubic foot, or ·284 lb. per cubic inch; a plate 1 foot square and 1 inch thick will therefore weigh very nearly 41 lbs.

The Admiralty requirements as regards steel castings for machinery are as follows:—

Tensile strength for pistons and intricate thin castings to be 30 to 37 tons per square inch, with an extension of at least 12 per cent. in 2 inches, and for ordinary castings 28 to 35 tons per square inch, with an extension of at least 15 per cent. in 2 inches.

Bars 1 inch square to bend cold without fracture, over 1⅝ inches radius:—for ordinary castings 90° for 28-ton steel, 60° for 35-ton steel, and other strengths in proportion; and for intricate thin castings 45° independent of strength.

Test pieces are to be taken from each casting.

All steel castings are also to stand being dropped from a height of about 12 feet upon a hard road or floor.

Lloyd's Rules for Quality and Testing of Steel Castings.

For purposes for which **cast iron** is ordinarily employed, such as propeller bosses and blades, bed-plates, engine framing and columns, brackets, weigh-shaft levers, pistons, cylinder covers, eccentric straps, bearing brushes, &c., the castings must be sound, and are to be subjected to such drop and hammering tests as below.

Section 5.—1. Process of Manufacture.—Steel for castings shall be made by the Open Hearth process, acid or basic, or by such other process as may be approved.

2. **Annealing.**—All steel castings shall be thoroughly annealed in a properly constructed annealing furnace, which must permit of the whole casting being uniformly raised in temperature throughout its whole extent to the necessary intensity. The casting shall be allowed to cool down prior to removal from the annealing furnace; and if subsequently heated for any purpose it shall again be similarly annealed if required by the Surveyor.

3. **Testing and Inspection.**—The following tests and inspections shall be made, but in the event of any casting proving unsatisfactory in the course of preparation or erection, such casting shall be rejected, notwithstanding any previous certificate of satisfactory testing.

4. **Tensile and Bend Test Pieces.**—The tensile strength and ductility shall be determined from standard test pieces, which are to be prepared from sample pieces on the casting. These sample pieces are not to be cut or partially cut from the castings until the annealing of such castings has been completed.

5. **Number of Tests.**—At least one tensile test and one cold bend test are to be taken from each casting. In castings of complex design,

referred to in paragraph 12, at least two tensile and two cold bend tests are to be taken. Where a casting is made from more than one charge of steel, at least four tensile and four cold bend tests are to be taken from pieces cast as far apart as possible on the casting, some test pieces being taken from as near the top and others from as near the bottom of the casting as practicable.

6. **Dimensions of Tensile Test Pieces.**—The tensile test pieces are to be turned to a diameter of ·564 inch with a gauge length of 2 inches, or a diameter of ·798 inch with a gauge length of 3 inches, or a diameter of ·977 inch with a gauge length of $3\frac{1}{2}$ inches.

7. **Dimensions of Bend Test Pieces.**—The bend test pieces are to be of a rectangular section, 1 inch wide by $\frac{3}{4}$ inch thick, with the edges rounded to a radius of $\frac{1}{16}$ of an inch. They are to be bent over the thinner section. The bending may be performed either by pressure or by blows.

8. **Tensile Tests.**—The tensile breaking strength determined from test pieces of standard dimensions is to be between the limits of 26 and 35 tons per square inch, with an elongation of not less than 20 per cent. measured on the standard test piece.

9. **Bend Tests.**—The bend test pieces must withstand without fracture being bent cold through an angle of 120 degrees, the internal radius of the bend being not greater than one inch.

10. **Additional Tests before Rejection.**—Should either the tensile or bend test or both fail and the Surveyor consider the fractured test piece or test pieces, or the results obtained therefrom, do not fairly represent the quality of the casting, a duplicate of the test or tests which failed shall be made if requested. In such cases the quality of the casting shall be judged by the result of the duplicate test or tests, and not by the original test or tests which failed.

11. **Percussive Tests.**—The castings are to be dropped on hard ground from a height of from 7 to 10 feet, according to the design, shape, and weight of the casting.

12. **Castings of Complex Design.**—Castings of complex design which would be liable to be deformed if submitted to the drop or percussive tests, may have this test dispensed with provided two tensile and two cold bend tests be made upon pieces taken from positions as far apart as possible on each casting; one tensile and one bend test being taken from as near the top and the others from as near the bottom of the casting as practicable.

13. **Hammering Tests.**—After being subjected to the percussive test, the casting in each case is to be subsequently slung up and well hammered with a sledge hammer not less in weight than 7 lbs., to satisfy the Surveyors that the casting is sound and without flaw. This hammering test is also to be applied to castings which may not have been submitted to a percussive test.

14. When the castings are to be used for purposes for which cast iron is ordinarily employed, they need not be submitted to tensile and bend tests, but they must be submitted to the drop and hammering tests specified in paragraphs 11 and 13.

15. **Branding.**—Every casting, after it has satisfactorily withstood the prescribed tests, shall be clearly and distinctly marked by the Society's Surveyor, indicating that the casting has complied with the Society's requirements.

BOARD OF TRADE TESTS FOR CAST STEEL AND MALLEABLE CAST IRON.

Testing of Cast-Steel and Malleable Cast-Iron Material.

Cast Steel.

63. *Cast-steel stems, sternposts, rudders, propeller shaft brackets, steering quadrants, crossheads or tillers, and other important castings which are subject to considerable stress and strain.*

Tensile and bending tests are to be made from these castings in the manner described in sections 133 to 134. The tensile strength should be in accordance with the limits specified in section 133. It is desirable that the percentage of elongation and the bending angle should not be less than those specified in sections 133 and 134 respectively for castings of superior quality, but, provided there are no unusual features in the design of the castings and the scantlings are sufficient, the lower limit of elongation and the bending angle specified will not be objected to.

The Surveyor should also carefully examine each casting, and take all practicable measures to satisfy himself that it is sound and free from flaws and defects.

64. *Steel castings other than those referred to above, when fitted in positions such that the failure of the castings may affect the safety of the vessel.*

(*a*) *Side-scuttle frames and plugs.*

In cases where side-scuttle frames and plugs are made of cast-steel, the Surveyors should be guided by the following instructions as to tests.

When side-scuttles are fitted above the vessel's upper deck, or when in such a position that their sills will be at least 10 feet above the centre of the freeboard disc, no tests will be required, unless the Surveyor has reason to doubt the strength of the material. When intended to be fitted in a lower position, one frame and also one plug should be selected at random out of each fifty, and at least one frame and one plug for every vessel so fitted, and tested to destruction by bending by means of blows from a hammer when cold and before being machined.

Side-scuttle frames of ordinary form should be capable of being bent to an angle of at least 20° without fracture.

Plugs should be capable of being bent through an angle of at least 40° without fracture.

Malleable Cast Iron.

65. Owing to the great differences in the qualities of this material as produced by different methods of manufacture, its employment in the construction of passenger steamships should be watched with care.

The surface of malleable iron castings should not be removed by machining to a greater extent than is absolutely necessary.

66. *Side-scuttle frames and plugs.*

Side-scuttle frames and plugs should be capable of being bent through an angle of at least 15° and 30° respectively without fracture.

67. *Rail Stanchions.*

Rail stanchions of malleable cast iron may be accepted for vessels intended to ply exclusively in smooth water or partially smooth water limits, provided the stanchions are capable of withstanding the following test:—

Stanchions should not be passed unless they are capable of being bent while cold at least 6 inches from the straight in a length of 36 inches, before fracture.

Malleable cast-iron rail stanchions are not sanctioned for use in sea-going vessels.

Steel Bars and Plates.

Mild steel has now almost entirely superseded wrought iron for bars and plates; even for ventilators, uptakes, chimneys, casings, &c., it is cheaper than iron of the quality to stand the necessary working, rolling, &c.

In steel plates rolled from ingots the direction in which the test pieces are cut from the plates (*i.e.* in the direction of rolling or across that direction) is found to have less influence on the strengths obtained than is the case with iron plates, the differences observed usually lying between 5 and 10 per cent.

The percentage of carbon in mild steel bars and plates (27 to 30 tons per square inch) is usually between ·15 and ·25 per cent.

Weight of steel bars and plates.—The average weight of steel of this description is about 490 lbs. per cubic foot, or ·284 lb. per cubic inch, and a plate 1 foot square and 1 inch thick weighs about 41 lbs.

Experience in handling mild steel has shown that work cannot safely be continued after the red colour of the heat has disappeared; when cold, the material will stand very severe treatment, and it is equally ductile at a full red heat, but below 700° C. it seems to be in a critical and at times brittle and unreliable condition, and should not be handled again until it has cooled below 400° C., or been re-heated.

Also, when a plate has been locally heated or worked, the internal stresses set up seem to be much more severe in effect than in the case of wrought iron, and to avoid all risk of cracks, the plates should be carefully annealed as soon as possible.

Table CXXXI.—Comparative Requirements for Steel Castings.

	Board of Trade.			British Standards Committee.				Lloyd's	British Corporation.	
Quality.	Superior.	—	—	A.	B.	C.	D.	—	—	Thin castings.
Tensile Strength.	26/35	35/40	26/35	35/40	26/35	26/35	No tensile or bending tests required.	26/35	26/35	26/35
Elongation.	20 %	15 %	15 %	15%	20 %	15 %		20 %	20 % for 26 tons. 15 % for 35 tons.	May be reduced to 15% and 10%
Angle of Bend.	120°	60°	90°	60°	120°	90°		120°	120°	120°
Other Tests Required.							Hammer test only.	Percussion test required, also hammering tests.	Drop test and hammering test.	May be exempted from drop test. Hammering test to be applied.
Remarks.	Compares with Quality "B" of the British Standards Committee.	Compares with Quality "A" of the British Standards Committee.	Compares with Quality "C" of the British Standards Committee.				No corresponding Quality in Board of Trade Requirements.	Compares with Board of Trade "Superior" and British Standards Committee Quality "B"	Elongation slightly altered from Quality "B" of British Standards Committee.	No corresponding quality in either Board of Trade, Lloyd's or British Standards Committee Requirements.

Steel Forgings.

Steel forgings are similar in most respects to bars and plates, but, of course, not quite so dense or so uniform in texture and strength.

The Admiralty require that all important steel forgings for machinery shall be made from ingots, and test pieces from all important ingots and forgings must satisfy the following conditions:—

Ultimate tensile strength to be between 30 and 35 tons per square inch, and extension not less than 27 per cent. in 2 inches.

Bars 1 inch square should bend cold, without fracture, through an angle of 180° over a radius not greater than ¼ inch.

The ultimate tensile strength of material for crank and propeller shafts is to lie between 28 and 32 tons per square inch, and the extension must be at least 30 per cent. in 2 inches. 30 per cent. of top end of ingot to be removed before forging, and at least 3 per cent. from bottom end after forging. Sectional area of body of forging is not to exceed one-sixth original sectional area of ingot.

The weight of steel forgings may be taken as about 487 lbs. per cubic foot, or ·282 lb. per cubic inch.

Whitworth's fluid compressed steel weighs about 495 lbs. per cubic foot.

Lloyd's Rules respecting Ingot Steel Forgings.

Section 6.—1. Process of Manufacture.—Ingot steel for forgings shall be made by the Open Hearth process, acid, or basic, or by such other process as may be approved by the Committee.

The forgings must be sound. They are to be made from sound ingots, and from all important forgings such as cranks and propeller shafts, connecting-rods, piston-rods, the forgings must be gradually and uniformly forged. The sectional area of the body of the forgings (as forged) shall not exceed one-fifth of the sectional area of the original ingot, and no part of the forging (as forged) shall have more than two-thirds of the sectional area of the original ingot.

2. **Annealing.**—As for steel castings (*vide* page 441).

3. **Testing and Inspection.**—As for steel castings (*vide* page 441).

4. **Tensile and Bend Test Pieces.**—The tensile strength and ductility shall be determined from standard test pieces which are to be prepared from sample pieces cut lengthwise from the forging from a part of not less sectional dimensions than the body of the forging. Such standard test pieces shall be machined from the sample pieces without forging down, and the sample pieces shall not be detached from the forging until the annealing of such forging has been completed.

5. **Number of Tests.**—At least one tensile and one cold bend test are to be taken from each forging. Where a number of articles are cut from one forging, one tensile and one cold bend test from this whole forging will be sufficient.

6. **Dimensions of Tensile Test Pieces.**—As for steel casting.

7. **Dimensions of Bend Test Pieces.**—The bend test pieces are to be machined to a rectangular section 1 inch wide by ¾ inch thick, with the edges rounded to a radius of $\frac{1}{16}$th of an inch. They are to be bent over the thinner section. The bending may be performed either by pressure or by blows.

8. **Tensile Tests.**—The tensile breaking strength determined from test pieces of standard dimensions is to be between the limits of 28 and 32 tons per square inch, with an elongation on the standard test piece of not less than 29 per cent. for 28-ton steel, and 25 per cent. for 32-ton steel, and in no case must the sum of the tensile breaking strength and corresponding elongation per cent. be less than 57.

9. **Bend Tests.**—The bend test pieces must withstand without fracture being bent cold through an angle of 180 degrees, the internal radius of the bend being not greater than ¼ inch.

10. **Additional Tests before Rejection.**—Should either the tensile or bend test, or both, fail, and the Surveyor consider the fractured test piece or test pieces, or the results obtained therefrom, do not fairly represent the quality of the forging, a duplicate of the test or tests which failed shall be made if requested by the maker. In such cases the quality of the forging shall be judged by the result of the duplicate test or tests and not by the original test or tests which failed.

11. **Branding.**—Every forging, after it has satisfactorily withstood the prescribed tests, shall be clearly and distinctly marked by the Society's Surveyor, indicating that the forging has complied with the Society's requirements.

12. **General.**—The requirements as to annealing and testing are intended to apply to shafts of all descriptions, also to connecting-rods and piston rods which require to be made in several heats. They are not intended to apply to small forgings which during their last stage of manufacture are uniformly heated throughout.

British Corporation Rules for Steel.

The British Corporation require the following:—

(i) Boiler shell plates and butt straps 28 to 32 tons tensile, with at least 20 per cent. elongation in 8 inches.

(ii) Plates for flanging 26 to 30 tons and 23 per cent. at least.

(iii) Plates and stays for furnaces and combustion chambers 26 to 30 tons and 23 per cent.

(iv) Rivet bars, same tensile but with 25 per cent. minimum extension or 31 per cent. with 3½ diameters.

Copper.

Copper in its unalloyed condition is hardly ever used. For pipes and for fire-box plates of loco type boilers 0·25 to 0·45 per cent. of arsenic is added.

The tenacity of such sheet copper is about 30,000 lbs. per sq. inch.

Annealed copper wire has a strength of 40,000 lbs. per sq. inch.

Wire of copper with 1 per cent. aluminium will stand 78,000 lbs.

per square inch, and French wire having a small addition of silicon to the copper as much as 128,000 lbs. Wire of copper with 0·529 per cent. of antimony stands 78,000 lbs.

Sheet copper 8·805 specific gravity has a weight equivalent to about 550 lbs. per cubic foot, or ·318 lb. per cubic inch. A square foot 1 inch thick weighs 45·83 lbs.

For weights of copper pipes see page 492.

The Admiralty specify that strips cut from steam and other pipes, either longitudinally or transversely, are to have an ultimate tensile strength of not less than 13 tons when annealed in water, and are to elongate at least 35 per cent. in 2 inches, or 30 per cent. in 4 inches.* They are also to bend through 180° until the two sides touch, and to stand hammering to a fine edge, when cold, without cracking.

Copper with 2 per cent. aluminium must stand 15 tons (33,600 lbs.) per square inch and show an extension of 40 per cent. This alloy weighs about 540 lbs. per cubic foot. An alloy of 92·5 copper and 7·35 aluminium has a very high resistance to alternating stresses, is tough and has high tensile strength.

Common Bronze or Gun-metal.†

Bronze is composed of copper and tin in various proportions, and a small percentage of zinc is usually added to ensure sound castings and permit of "tooling" easily.

Its strength depends mainly on the proportions and quality of the metals forming its composition, but is much affected by such circumstances as the size of the casting, the rate of cooling, and the skill of the founder in the heat treatment and mixing them, ventilating the moulds, relieving the cores, &c.

In large castings which cool slowly there is a great tendency for the metals to separate from one another to some extent, and the average strength is therefore usually less than in small castings; as a general rule the more quickly the casting is cooled the stronger the metal is.

The metal sets very rapidly, and contracts nearly $\frac{5}{16}$ inch per foot on an average, so that in large castings the cores must be very quickly relieved if the casting is not to be drawn and porous and of low tenacity.

With a mixture of 90 per cent. copper and 10 per cent. tin, a carefully made test bar may be got to show a strength of nearly 17 tons (38,000 lbs.) per square inch. With 84 per cent. copper and 16 per cent. tin, a much harder metal is obtained (the hardness of gun-metal varies almost directly with the percentage of tin in the mixture), with an ultimate strength of test piece of about 16 tons (35,000 lbs.) per square inch.

For heavy bearings, 79 per cent. copper and 21 per cent. tin is sometimes used; the resulting metal is very hard, and test pieces show a strength of 13½ to 14 tons (30,000 to 31,000 lbs.) per square inch.

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* B.M.E.D. & C.C. require 14 tons with elongation of 40 per cent. in 2 inches.

† Guns are no longer made of bronze, but now invariably of steel.

For bearings the metal is improved by the addition of a small amount of lead ; a little zinc is always necessary to facilitate machining.

Admiralty Bronze.—For all ordinary castings and steam fittings in connection with the machinery, the alloy used must contain *not less* than 10 per cent. of tin, and *not more* than 2 per cent. of zinc. For air-compressing machinery and torpedo fittings, &c. (where the working pressure is over 3000 lbs. per square inch), the mixture specified is:—Copper, *not less* than 86 per cent.; tin, *not less than* 10 *nor more than* 12 per cent. ; zinc, *not more* than 2 per cent.

The ultimate tensile strength of bronze is to be not less than 14 tons (31,360 lbs.) per square inch, and it must extend at least 7½ per cent. in 2 inches before breaking.

Fairly good ordinary bronze ought to show a strength of about 12 tons (27,000 lbs.) per square inch, and should extend 10 per cent. in a length of 2 inches before breaking.

Its weight is about 545 lbs. per cubic foot, or ·315 lb. per cubic inch.

Phosphor Bronze.

This metal is composed of copper and tin, with a small proportion (about ½ per cent.) of phosphorus. It is harder than ordinary gun-metal, very close-grained, and of superior strength. The average ultimate strength is about 15½ tons (35,000 lbs.) per square inch, while that of some grades of the metal is as high as 22 tons; it is, however, "red-short," so that when heated it is liable to crack. Great care is required in melting and running it, and repeated meltings very much reduce its virtue, as the phosphorus disappears. Sheet phosphor bronze and rods, wire, &c., are most useful, and this metal is now often used for the blades of turbines. The elastic limit of phosphor bronze is very high.

Manganese Bronze.

Manganese bronze now consists of copper, tin, and zinc (according to the grade of metal required), with the addition of a proportion of ferro-manganese.

The weight of this material is about 535 lbs. per cubic foot, or ·31 lb. per cubic inch.

Rolled rods of manganese bronze can be obtained of strengths varying from 28 to 32 tons (63,000 to 72,000 lbs.) per square inch, and showing an elongation of 40 per cent. to 15 per cent in 2 inches.

Zinc Bronzes.

Parsons' Manganese Bronze Co. were the first to manufacture manganese bronze for propellers, &c.; their alloy formerly had a very considerable content of manganese and possessed a high tensile strength, &c. It has of late years been replaced by alloys having much less manganese, but some other metals in small quantities, which, with the methods of

melting and treating followed by the company, possess quite as high and often higher strength, with greater toughness, and safer for marine castings. The following are the chief products of this company:—

Crotarite, suitable for boiler stays and other fittings exposed to heat. The melting point is nearly as high as copper (2000° F.), it is highly malleable, and has a tensile strength of 25 to 26 tons, with 30 to 40 per cent. elongation. Elastic limit is 16 tons.

Immadium, suitable for shafts, pumps, rods, &c., exposed to sea-water, being incorrodible. Mild variety has an elastic limit of 18 tons and an ultimate tensile of 36 tons, with 25 per cent. elongation. In the cast state it is very suitable for boiler mountings and fittings, as it has an elastic limit of 9 tons at 500° F. and an ultimate of 23·5 tons, with 22·5 per cent. elongation.

Turbadium is a special bronze for high-speed propellers, being capable of resisting the corrosion of sea-water and erosion due to high velocity. The elastic limit is 18 tons and the ultimate tensile 40 tons, with an extension of 18·5 per cent. in 8 inches. This can be wrought at a cherry-red heat.

White Brass, as a bearing metal, is another product of this company.

J. Stone & Co.'s Bronzes.—This firm supply several kinds of copper alloys suitable for the different parts of marine machinery, the principal of which are:—

1. *Toughened Copper*, suitable for boiler stays and other fittings which require to be tough and strong when at a fairly high temperature. At 60° F. its ultimate tensile is 18·1 tons per square inch, with an elongation of 55 per cent. in 2 inches; while at 400° F. it is still 55 per cent., with 14·1 tons ultimate and 7·4 tons elastic limit. At 800° F. the elastic limit is 8·7 tons, with an elongation at fracture of 37 per cent.

2. *Special Manganese Bronze*, for propellers of high revolution, is guaranteed to have an ultimate tensile of 32 tons, with an elongation of 20 per cent. in 2 inches.

3. *Patent Bronze* (*No. 4 quality*) when in cold rolled bars have an ultimate strength of 45 tons, with 10 per cent. elongation in 2 inches and an elastic limit of 37·5 tons.

In the annealed state the ultimate is 34 tons and 36 per cent., with the elastic limit at 17 tons.

In normal state it withstands 40 tons per square inch compression, with a shortening of only ·084 per inch, with a bulge of ·035 in a diameter of ¾ inch.

4. *Stone's Ordinary Propeller Bronze.*—Six Admiralty tests gave an average of 33·8 tons ultimate tensile, with an elongation of 23·5 per cent. in 2 inches.

5. *White Bronze* for bearings, &c., is composed largely of tin.

Bull's Metal Company manufacture another of the zinc bronzes for propellers, which has an ultimate tenacity of 31 tons, with an elongation of 25 per cent.; its elastic limit is 13·2 tons. When rolled the tensile is 34 tons, with an elongation of 34 per cent. in 2 inches and an elastic limit of 26·8 tons. When this metal is at a temperature of 400° F.,

the ultimate elongation is 14·6 per cent., while the elastic limit is 21 tons.

Melloid.—A special alloy supplied by this company to resist heat, has a tensile of 98,000 at 60° F., while at 600° F. it is as high as 79,000 lbs. or 35 tons, with an elongation of 18 per cent. in 2 inches; and even at 800° F. it has an ultimate tenacity of 31,800 lbs., with an elongation of 50 per cent.

Brass.

Brass is essentially an alloy of copper and zinc only, of which there are many variations in composition. The most important in marine engineering are:—

1. *Muntz Metal*, an alloy of 59 to 60 per cent. of copper with 41 to 40 of zinc. It should have a tensile strength of at least 22 tons, with an elongation of 27 per cent.; when rolled it has an elastic limit of 36,000 lbs. It is very ductile and can be forged hot. Weight 512 lbs. per cubic foot, 0·296 per cubic inch.

2. *Condenser Tube Metal* is usually made with 70 per cent. copper and 30 zinc, and should in the drawn state have a tensile strength of 36 tons ultimate. Admiralty condenser tubes are made of 70 per cent. best selected copper, 29 of silesian zinc, and 1 of tin.

3. Condenser tubes of a mixture of 70 copper, 28 zinc, and 2 lead are said to resist corrosion even better than the Admiralty mixture.

4. *Naval Brass* is virtually Muntz metal with an addition of 1 per cent. of tin. It is generally composed of 62 copper, 37 zinc, and 1 tin; and in the wrought state should have an ultimate tensile strength of 60,800 lbs., with an elongation of 19 per cent., and the elastic limit should be 42,400 lbs. It can be forged, &c., and withstands the action of sea-water better than Muntz metal.

The Admiralty require naval brass rods of ¾ inch and upwards to have an ultimate tensile of 22 tons and an extension *before* fracture of at least 10 per cent. in 2 inches; they must also bend through 75° with a radius of one diameter; and when hot are to be forged to a fine point.

Yellow Brass, for ornamental and ordinary castings where strength is not wanted, is composed of 2 copper to 1 zinc. The addition of a little lead improves the colour and facilitates machining. Its ultimate tensile is about 11 tons.

Aluminium is now very extensively used in the arts, and from its lightness would be a most useful one for marine engineers if it did not become so quickly and seriously affected by sea-water. In the pure state it is practically of no use to them, but when alloyed its tensile strength and durability are much improved. It is now so cheap that it can be used for any purpose so far as cost is concerned, and is used for telegraph wire in place of copper in certain localities, especially since copper has been so dear (£72 per ton). Its conductivity is good; its resistance is 2·839 per c.c. as against 1·621 copper wire and 9·637 iron. Its ultimate tensile in wire form is 8 tons. When alloyed with

6 per cent. of copper and rolled into sheets, rods, &c., it has a strength of 11 to 12 tons after, and 14 to 16 tons before annealing.

Duralumin is a 90 per cent. alloy of aluminium made by Vickers Limited, having a specific gravity of only 2·8, a melting point of 1200° F., an ultimate tensile as high as 40 tons, when the extension, however, is small; at 28 to 30 tons ultimate, the extension is 15 per cent. in 2 inches, while with 25 tons it is 20 per cent.

Monel Metal is a natural nickel bronze, having a very high tensile strength both in the cast and rolled state. Castings have an ultimate tensile from 33·7 to 36·5 tons, with an elongation of 38 to 28·5 per cent.; the elastic limit being 16 to 18 tons per square inch in 2 inches.

Rolled bars, plates, &c., have an elastic limit of 25 tons and an ultimate of 42 tons, with an elongation of 41 per cent. Can be forged hot and brazed. This alloy is uncorrodible and used for screw propellers of high-speed craft in the U.S.A. Navy. Its specific gravity is 8·87; weight per cubic inch 0·319 lbs. cast, 0·323 rolled; melting point 2480° F.

The Delta Metal Co. produces several varieties of zinc bronze and other alloys.

D. M. No. 1, of fine golden colour, very hard, resists corrosion, and takes place of steel. As cast, it has an ultimate strength of nearly 41 tons, with an elongation of 20 per cent. At a red heat it is highly malleable. When wrought, the ultimate tensile is 43·27 tons, with an elongation of 27 per cent.

This metal is used for extrusion into bars of various sections, the ultimate strength of which is nearly 50 tons, with 26 per cent. elongation and 24·5 tons yield point.

It has high resistance to corrosion from action of sea water and acids. A shearing test showed its resistance to be no less than 23·11 tons per square inch.

D. M. No. 2 is a silver white metal, malleable, takes a high polish, and therefore suitable for fittings, &c. When extruded it has a yield point of 22 tons, and the ultimate strength 42·45 tons with 14 per cent. of elongation. As cast, it has an ultimate of 38 tons with 10 per cent.

D. M. No. 3 is suitable for condenser tubes and pipes generally, as it can be easily drawn and resists corrosion.

D. M. No. 4 is equal to steel in strength and toughness, strongly resists corrosive effects of sea water and acids, chemical gases, &c., therefore largely used in chemical engineering as well as marine. It is used also in the cast form, and bars cut from chill castings gave 19 tons limit of elasticity, 26·8 ultimate with an elongation of 36·8 per cent.

It can be forged or drawn into tubes when the ultimate is 34·4 tons with 26·25 per cent. elongation. It can also be stamped with equally good results.

It is often used in sheets for boat-building and similar purposes, and the company supplies for this purpose sectional bars, angles, zeds, channels, &c. &c.

D. M. No. 5 is a bearing metal.

D. M. No. 7 is suitable for high temperature.

Table CXXXII.—Composition of White (Bearing) Metals.

Name of Metal.	Cu.	Sn.	Sb.	Zn.	Pb.	Fe.	Miscellaneous.
Dewrance's Locomotive,	22·2	33·3	44·4	...	...	...	...
For gland packings, .	7·8	88·1	3·5	...	2·6	...	...
Used in the German Navy, . . .	7·5	85·0	7·5	...	...	...	...
,, French Navy,	7·0	7·5	...	78·5	7·0	...	...
,, British Navy,	5·5	86·0	8·5	...	...	...	...
Babbit's Metal, . .	8·5	83·0	8·5	...	...	...	...
Fenton's ,, . .	4·4	16·6	...	79·0	...	...	...
Magnolia, ,, . .	...	...	21·0	...	78·0	1·0	...
,, ,, . .	...	4·6	13·0	...	82·0	...	0·4
Kingston's Metal, .	6·0	88·0	...	...	...	...	6 Hg
Parson's white brass, .	5·6	17·5	0·8	76·1	...	...	...
,, ,, metal, .	...	58·5	2·0	39·5	...	...	...
,, ,, ,, .	1·0	68·0	...	30·5	0·5	...	...
For common bearings, .	10·0	...	10·0	...	80·0	...	...
,, heavily loaded bearings, . . .	64·0	5·0	...	...	30·0	...	1 Ni
Plumtine, . . .	0·5	40·5	11·0	...	48·5	...	...

A specially good white metal for heavy bearings is made by mixing 6 parts of tin with 1 of copper, and 6 parts of tin with 1 of antimony, and then adding the two mixtures together.

The exact Admiralty specification is,—at least 85 per cent. tin, not less than 8 per cent. antimony, and about 5 per cent. copper; zinc or lead not to be used. This is somewhat harder than the above.

Table CXXXIII.—Properties of Metals.

Metal or Alloy.	Ultimate Strength.	Extension in 8 inches.	Elastic Limit.	Compressed Strength.	Shear per cent. of Tensile.	Specific Gravity.	Weight of		Melts at ° F.	Conducts Heat. Silver 100	Extension per 10° F.
							Cubic Inch.	Sq. Ft. 1 inch Thick.			
	Lbs.	Per cent.	Lbs.	Lbs.							
Cast iron, good castings,	25,000	...	...	110,000	162·0	7·20	0·260	37·50	2100	...	...
Wrought iron, good boiler quality,	50,000	12·0	26,000	...	80·0	7·68	0·280	40·0	2800	11·90	·0000670
,, ,, bar ,,	60,000	15·0	30,000	...	...	7·62	0·276	39·70	,,	,,	,,
,, ship plate ,,	45,000	10·0	27,500	...	...	7·68	0·280	40·0	,,	,,	,,
Steel castings, good mild,	63,000	15·0	34,000	...	...	...	...	..	2500	,,	,,
,, wrought (0·15 carbon),	58,000	25·0	29,000	...	80·0	7·86	0·284	40·94	,,	,,	,,
,, ,, (0·40 ,,),	78,000	22·0	33,000	...	...	...	...	..	...	...	..
,, ,, (,, ,,), oil tempered,	90,000	...	55,000	...	...	..	...	....	...	...	...
Nickel steel (3½ Ni, 0·29 C), forged,	85,000	30·0	54,000	...	...	...	...	...	...	...	...
,, (3½ Ni), oil tempered,	90,000	...	60,000	...	...	...	...	...	...	...	...
,, mild quality boiler plate,	74,000	...	43,000	...	...	...	...	...	...	...	...
,, high tensile ,,	114,000	...	64,000	...	...	...	...	...	...	...	...
Chrome steel (2·5 Cr, 0·39 C), forged,	65,400	...	52,600	...	...	...	..	....	...	...	...
,, (,, ,,), unannealed,	120,900	...	97,400	...	...	...	...	...	...	...	...
Vanadium steel (0·257, V),	88,000	24·0	76,300	...	...	...	...	...	...	...	...
Chrome vanadium steel (1·0 Cr + ·15 V.),	117,800	25·0	77,000	...	...	...	...	...	...	...	...
Manganese steel (11·0 Mn), forged,	134,000	35·0	...	..	...	...	...	...	...	...	...
Nickel, cast,	36,400	...	24,600	...	...	8·30	...	...	2820	...	...
,, forged,	72,100	...	25,500	...	...	8·80	0·318	45·80	2820	...	·0000705
German silver (19 Ni), rolled,	...	...	...	...	...	8·80	0·318	45·80	...	...	...
Copper, annealed sheets,	30,000	...	5,600	...	86·7	8·85	0·320	46·10	2000	73·6	·0000930
,, bars	36,000	...	...	...	...	,,	...	...	,,	,,	,,
Zinc, cast,	3,000	...	...	..	...	7·00	0·253	36·42	800	28·1	·0001620
,, rolled,	7,500	...	...	...	...	,,	...	...	,,	,,	,,
Tin, cast,	4,700	...	...	...	...	7·30	0·264	38·00	440	15·2	·0001280
Muntz metal, rolled,	56,000	27·0	36,000	...	..	8·20	0·296	42·72	1630	...	...
Naval brass	60,800	19·0	42,400	...	55·8	8·20	0·296	42·72	1640	...	...
Cobalt (cast annealed)	36,000			12,200		8·81					

Table CXXXIII.—Properties of Metals—*continued.*

Metal or Alloy.	Ultimate Strength.	Extension in 2 inches.	Elastic Limit.	Compressed Strength.	Shear per cent. of Tensile.	Specific Gravity.	Weight of		Melts at ° F.	Conducts Heat. Silver 100	Extension per 10° F.
							Cubic Inch.	Sq. Ft. 1 inch Thick.			
	Lbs.	Per cent.	Lbs.	Lbs.							
Manganese bronze (Parsons), cast	No. 2, 67,000	24·5	33,000	...	...	8·51	0·308	44·34	1660	...	...
,, ,, rolled, annealed,	67,500	26·2	33,000	...	...	8·40	0·304	43·76	,,	...	..
Stone's bronze, cast,	77,500	...	38,200	127,000	...	..	...	...	...	...	...
,, rolled,	80,600	..	66,000	...	...	...	...	...	...	...	...
Delta metal, rolled,	89,000	...	...	...	...	8·40	0·304	43·76	1800	...	...
Bull's metal, cast,	70,000	...	29,600	176,000	...	8·30	0·300	43·24	1800	...	...
,, rolled,	78,000	34·0	60,000	224,000	...	,,	,,	,,	,,	...	...
Aich's metal, ,,	57,300	25·6	...	...	...	...	...	...	...	...	...
Phosphor bronze (wheel B), cast,	39,200	...	38,900	...	113·0	8·60	0·311	44·80	1900	...	...
,, ,, rolled,	68,500	...	62,000	...	63·0	,,	,,	,,	,,	...	...
Silicium bronze wire,	105,000	...	...	...	...	...	...	...	...	...	...
Admiralty bronze, cast	30,000	...	17,000	...	...	8·66	0·325	46·80	1700	...	...
Aluminium (pure comml.), cast,	11,800	...	...	...	...	2·56	0·092	13·33	1160	31·33	·0000280
,, (,,), bar, rolled,	28,200	...	19,300	...	...	2·70	0·098	14·06	,,	,,	,,
,, (,,), plates, rolled,	16,500	...	9,600	...	...	2·70	0·098	14·06	,,	,,	,,
,, (6·0 Cu), plates rolled,	34,400	3·7	30,600	...	...	3·07	0·111	16·00	...	...	...
Aluminium bronze (10 Al), rolled	100,000	22·0	...	...	...	7·65	0·277	39·86	1750	...	...
,, (5 Al), cast,	56,000	50·0	...	...	...	...	...	...	...	...	...
Antimony, cast,	...	...	...	..	...	6·71	0·243	35·00	1166	4·03	...
Bismuth, ,,	...	...	...	...	...	9·80	0·355	51·05	514	1·80	..
Lead, sheet,	3,300	...	...	...	...	11·37	0·411	59·24	620	8·50	...
Manganese, cast,	...	...	...	...	...	8·01	0·289	41·52	3450	...	...
Platinum, rolled,	...	...	...	...	...	21·50	0·778	112·0	3225	8·40	...
Silver, rolled,	...	...	...	...	...	10·53	0·381	54·86	1730	100·0	...
Parsons' immadium, cast,	80,600	15·0	36,100	...	...	...	...	...	...	...	...
,, ,, forged,	90,000	20·0	45,000	...	...	...	...	...	...	...	...
,, turbadium, cast,	90,200	18·58	40,320	...	...	...	...	...	...	...	...
Monel metal, cast,	78,500	33·0	38,100	...	...	8·87	0·319	46·0	2480	17·0	·000096

Table CXXXIV.—Prices of Materials.*

Materials at Maker's Works.	Prices, 1914.	August 1911.
Pig iron, Shropshire, cold blast, No. 3,	£6 5 0 ℔ ton.	£5 5 0
Pig iron, Hæmatite, West Coast, No. 3,	3 5 0 ,,	3 8 0
Pig iron, Hæmatite, East Coast, No. 3,	3 3 0 ,,	3 3 0
Pig iron, Scotch, good brand, No. 3,	2 16 0 ,,	2 18 0
Pig iron, Scotch, good brand, No. 1,	3 10 0 ,,	3 3 0
Pig iron, Cleveland, good brand, No. 3,	2 10 0 ,,	2 7 0
Pig iron, Cleveland, good brand, No. 1,	2 12 6 ,,	2 10 6
Pig iron, Lincoln, good brand, Foundry,	2 13 0 ,,	2 8 6
Pig iron, Derby, Leicester, good brand, Foundry,	2 18 0 ,,	2 10 0
Wrought iron, Cleveland bars, ordinary,	7 10 0 ,,	6 10 0
Wrought iron, Cleveland bars, best,	7 17 6 ,,	7 0 0
Wrought iron, Cleveland ship plates,	6 15 0 ,,	6 15 0
Wrought iron, Cleveland ship angles,	7 10 0 ,,	6 15 0
Wrought iron, Staffordshire best bar,	7 0 0 ,,	6 17 6
Wrought iron sheets,	9 5 0 ,,	7 12 6
Steel ,, ,,	8 5 0 ,,	
,, angles and tees, ship quality,	6 2 6 ,,	6 7 6
Steel plates, ship quality,	6 10 0 ,,	6 15 0
,, boiler plates, basis,	7 15 0 ,,	7 10 0
,, forgings, plain shafts with couplings,	£20 to 25 0 0 ,,	£10 to £15
Steel for tools, common,	4d. to 6d. ℔ lb.	4d. to 6d.
,, for tools, good,	6d. to 9d. ,,	6d. to 9d.
,, for tools, self-hardening, for high-speed,	£0 1 6 ,,	£0 1 8
Steel for tools, very best, for high speed,	0 2 0 ,,	2s. 3d. to 2s. 6d.
Copper ingots, G.M.B.,	65 5 0 ℔ ton.	£56 12 6 ℔ ton.
,, ,, tough,	71 5 0 ,,	60 5 0 ,,

* Now altogether abnormal.

Table CXXXIV.—Prices of Materials—(*continued*).

Materials at Maker's Works.	Prices, 1914.	August 1911.
Copper ingots, best selected,	£71 5 0 ℔ ton.	£60 5 0
,, locomotive plates, .	83 0 0 ,,	73 0 0
,, sheets, basis, . .	81 0 0 ,,	71 0 0
,, pipes, solid drawn, 1¼ in. to 3 in., . .	0 0 10¼ ℔ lb.	0 0 8½
Copper pipes, solid drawn, under 1¼ in. and over 3 in.,	0 0 10¼ ,,	0 0 8¾
Copper pipes, brazed, .	0 0 8¾ ,,	0 0 8¼
Brass condenser plates, Mercantile, . . .	0 0 7½ ,,	0 0 6¼
Brass condenser plates, Naval, ...	0 0 8 ,,	0 0 6¾
Brass tubes, 66% copper, .	0 0 8⅜ ,,	0 0 6¾
Brass condenser tubes, 70% copper, Mercantile, .	0 0 8⅜ ,,	0 0 6¾
Brass condenser tubes, 70% copper, Admiralty, .	0 0 8⅞ ,,	0 0 7¼
Tin English ingots, . .	170 10 0 ℔ ton.	190 0 0
Zinc slabs, Silesian, . .	21 11 0 ,,	26 10 0
,, ,, best, .	22 7 6 ,,	27 5 0
Aluminium ingots, Nos. 1, 2, and 4, . . .	95 0 0 ,,	
Aluminium ingots, No. 6,	88 0 0 ,,	61 0 0
,, sheets, from Nos. 1 to 30, wire gauge,	£130 to 150 0 0 ,,	
Lead pig, ...	18 10 0 ,,	14 0 0
,, pipe, . . .	£14 10s. to 15 0 0 ,,	16 2 6
Nickel ingots, of good q'l'ty,	0 1 6½ ℔ lb.	
Muntz metal sheet, . .	0 0 8½ ,,	0 0 6¼
Naval brass rods, . .	0 0 8¾ ,,	0 0 6½
Phosphor bronze ingots, for general castings, .	98 0 0 ℔ ton.	£88 to £92
Phosphor bronze ingots, suitable for bearings, .	104 0 0 ,,	(basis).
Phosphor bronze rods, .	0 1 1½ ℔ lb.	0 1 1¾
,, sheet, .	0 1 2 ,,	0 1 0½
Manganese bronze ingots, .	£70 to 80 0 0 ℔ ton.	
Antimony cakes, . .	28 0 0 ,,	£28 2 6
Bismuth, ...	0 8 3 ℔ lb.	
Platinum wire, . . .	5 5 0 ℔ oz. troy.	
,, sheets, . .	5 12 0 ,,	
Copper wire, . . .	0 0 9 ℔ lb.	
Steel ,, . . .	0 0 1 ,,	
Aluminium wire, . .	0 1 0 ,,	

PLATES OBTAINABLE IN GREAT BRITAIN.

Area List for Sheared Steel Plates.

Maximum dimensions of plates rolled, *except by special arrangement.*

Rectangular Plates. Maximum.					Circular Plates. Maximum.		
Thickness.	Length.		Width.		Area in feet.	Thickness.	Diameter in feet.
ins.	ft.	ins.	ft.	ins.	ft.	ins.	ft. ins.
1/4	30	0	6	0	120	1/4	6 6
5/16	30	0	6	0	132	5/16	7 0
3/8	35	0	8	0	190	3/8	9 0
7/16	40	0	8	0	200	7/16	9 6
1/2	40	0	10	0	240	1/2	11 0
9/16	50	0	10	0	260	9/16	11 0
5/8	50	0	11	0	300	5/8	12 0
3/4	50	0	12	6	325	3/4	13 0
7/8	50	0	12	6	350	7/8	13 0
1	50	0	12	6	430	1	13 0
1 1/8	50	0	12	6	450	1 1/8	13 0
1 9/16	50	0	12	6	430	1 3/16	13 0
1 1/4	50	0	12	6	415	1 1/4	13 0
1 3/8	50	0	12	6	380	1 3/8	13 0
1 1/2	50	0	12	6	340	1 1/2	13 0
1 5/8	50	0	12	6	300	1 5/8	13 0

Area List for Thick Plates, which can be Planed to Sizes.

Thickness.	Maximum Length.		Maximum Width.		Maximum Area.
ins.	ft.	ins.	ft.	ins.	ft.
1 3/4	30	0	12	0	288
2	25	0	12	0	250
2 1/2	25	0	12	0	200
3	25	0	12	0	160
4	25	0	11	0	120
5	20	0	10	0	100
6	20	0	9	0	81

The area divided by the length gives the width of a plate that can be rolled in any given thickness, and the area divided by the width gives the length.

STEEL BARS OBTAINABLE IN GREAT BRITAIN.

Section List of Bars.

FLATS.

Width. ins.		Thickness. in.		ins.
$\frac{1}{2}$	×	$\frac{1}{8}$	to	$\frac{3}{8}$
$\frac{5}{8}$	×	$\frac{1}{8}$	,,	$\frac{1}{2}$
$1\frac{1}{16}$	×	$\frac{1}{8}$	,,	$\frac{5}{8}$
$\frac{11}{16}$	×	$\frac{1}{8}$	,,	$\frac{5}{8}$
$1\frac{3}{16}$	×	$\frac{1}{8}$	,,	$\frac{5}{8}$
$\frac{7}{8}$	×	$\frac{1}{8}$	,,	$\frac{3}{4}$
1	×	$\frac{1}{4}$	,,	$\frac{3}{4}$
$1\frac{1}{8}$	×	$\frac{1}{4}$	,,	1
$1\frac{1}{4}$	×	$\frac{1}{4}$	,,	1
$1\frac{3}{8}$	×	$\frac{1}{4}$	,,	1
$1\frac{1}{2}$	×	$\frac{1}{4}$	,,	1
$1\frac{5}{8}$	×	$\frac{1}{4}$	,,	$1\frac{1}{2}$
$1\frac{3}{4}$	×	$\frac{1}{4}$	,,	$1\frac{1}{2}$
$1\frac{7}{8}$	×	$\frac{1}{4}$	,,	$1\frac{3}{4}$

Width. ins.		Thickness. in.		ins.
2	×	$\frac{1}{4}$	to	$1\frac{3}{4}$
$2\frac{1}{4}$	×	$\frac{1}{4}$	,,	2
$2\frac{3}{8}$	×	$\frac{1}{4}$	,,	2
$2\frac{1}{2}$	×	$\frac{1}{4}$	,,	$2\frac{1}{4}$
$2\frac{3}{4}$	×	$\frac{1}{4}$	,,	$2\frac{1}{2}$
$2\frac{7}{8}$	×	$\frac{1}{4}$	,,	$2\frac{1}{2}$
3	×	$\frac{1}{4}$	,,	$2\frac{3}{4}$
$3\frac{1}{4}$	×	$\frac{1}{4}$	,,	$2\frac{3}{4}$
$3\frac{1}{2}$	×	$\frac{1}{4}$	,,	3
$3\frac{3}{4}$	×	$\frac{1}{4}$	,,	3
4	×	$\frac{1}{4}$	,,	3
$4\frac{1}{4}$	×	$\frac{3}{8}$	,,	3
$4\frac{1}{2}$	×	$\frac{3}{8}$	,,	3
$4\frac{3}{4}$	×	$\frac{3}{8}$	,,	3

Width. ins.		Thickness. ins.		ins.
5	×	$\frac{3}{8}$	to	4
$5\frac{1}{4}$	×	$\frac{3}{8}$	,,	4
$5\frac{1}{2}$	×	$\frac{3}{8}$	,,	4
$5\frac{3}{4}$	×	$\frac{3}{8}$	,,	4
6	×	$\frac{3}{8}$	,,	4
$6\frac{1}{4}$	×	$\frac{3}{8}$	,,	4
$6\frac{1}{2}$	×	$\frac{3}{8}$	,,	4
$6\frac{3}{4}$	×	$\frac{3}{8}$	,,	4
7	×	$\frac{3}{8}$	,,	4
$7\frac{1}{4}$	×	$\frac{3}{8}$	,,	4
$7\frac{1}{2}$	×	$\frac{3}{8}$	,,	4
$7\frac{3}{4}$	×	$2\frac{1}{4}$	,,	4
8	×	$2\frac{1}{4}$	,,	4

ROUND EDGE FLATS.

From 1 to 2 inches wide × $\frac{1}{8}$ to $\frac{5}{8}$ inch thick.

SQUARES.

ins.	ins.	ins.
$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$
$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$
$\frac{5}{8}$	$1\frac{1}{16}$	$\frac{3}{4}$
$1\frac{3}{16}$	$\frac{7}{8}$	$1\frac{5}{16}$
1	$1\frac{1}{8}$	$1\frac{1}{4}$
$1\frac{3}{8}$	$1\frac{1}{2}$	$1\frac{5}{8}$
$1\frac{7}{8}$	$1\frac{7}{8}$	2
$2\frac{1}{4}$	$2\frac{1}{4}$	$2\frac{3}{8}$
$2\frac{1}{2}$	$2\frac{5}{8}$	$2\frac{3}{4}$
$2\frac{7}{8}$	3	$3\frac{1}{8}$
$3\frac{1}{4}$	$3\frac{1}{2}$	$3\frac{3}{4}$
4	$4\frac{1}{4}$	$4\frac{1}{2}$
$4\frac{3}{4}$	5	$5\frac{1}{4}$
$5\frac{1}{2}$	$5\frac{3}{4}$	6

ROUNDS.

ins.	ins.	ins.	ins.
$\frac{1}{4}$	$\frac{9}{32}$	$\frac{5}{16}$	$\frac{11}{32}$
$\frac{3}{8}$	$\frac{13}{32}$	$\frac{7}{16}$	$\frac{15}{32}$
$\frac{31}{64}$	$\frac{1}{2}$	$\frac{17}{32}$	$\frac{9}{16}$
$\frac{19}{32}$	$\frac{5}{8}$	$\frac{21}{32}$	$\frac{11}{16}$
$\frac{25}{32}$	$\frac{25}{32}$	$\frac{25}{32}$	$\frac{13}{16}$
$\frac{27}{32}$	$\frac{7}{8}$	$\frac{29}{32}$	$\frac{15}{16}$
1	$1\frac{1}{32}$	$1\frac{1}{16}$	$1\frac{3}{32}$
$1\frac{1}{8}$	$1\frac{3}{16}$	$1\frac{1}{4}$	$1\frac{5}{16}$
$1\frac{3}{8}$	$1\frac{7}{16}$	$1\frac{1}{2}$	$1\frac{5}{8}$
$1\frac{5}{8}$	$1\frac{7}{8}$	2	$2\frac{1}{8}$
$2\frac{1}{4}$	$2\frac{3}{8}$	$2\frac{1}{2}$	$2\frac{5}{8}$
$2\frac{3}{4}$	$2\frac{7}{8}$	3	$3\frac{1}{8}$
$3\frac{1}{4}$	$3\frac{3}{8}$	$3\frac{1}{2}$	$3\frac{5}{8}$
$3\frac{3}{4}$	4	$4\frac{1}{4}$	$4\frac{1}{2}$
$4\frac{3}{4}$	5	$5\frac{1}{4}$	$5\frac{1}{2}$
$5\frac{3}{4}$	6	$6\frac{1}{4}$	$6\frac{1}{2}$
$6\frac{3}{4}$	7	$7\frac{1}{2}$	8
$8\frac{1}{2}$	9		

SLABS.

15	ins.	to	20	ins.	wide	×	5	ins.	to	8	ins. thick.
20	,,	,,	36	,,	,,	×	5	,,	,,	12	,, ,,
36	,,	,,	56	,,	,,	×	9	,,	,,	12	,, ,,

MAKER'S EXTRAS ON SIEMENS' STEEL BOILER PLATES AND BARS.

Marine, Land, and Locomotive Boiler Plates.

For every 3 inches or part over free limits in undernoted Table, 2s. 6d. per ton.

For every 3 inches or part over Width indicated in this Table, 2s. 6d. per ton.

Thickness of plate .	ins. $\frac{3}{8}$ and under $\frac{7}{16}$	ins. $\frac{7}{16}$ and under $\frac{1}{2}$	ins. $\frac{1}{2}$ and under $\frac{9}{16}$	ins. $\frac{9}{16}$ and under $\frac{5}{8}$	ins. $\frac{5}{8}$ and under $\frac{3}{4}$	ins. $\frac{3}{4}$ and under $\frac{7}{8}$	ins. $\frac{7}{8}$ and under 1	ins. 1 and under $1\frac{1}{4}$	ins. $1\frac{1}{4}$ and under $1\frac{1}{2}$	$1\frac{1}{2}$
Width supplied free of extra for width .	75	84	90	96	96	96	96	96	96	96

Weight over 80 cwts., 5s. per 5 cwts. or part up to 120 cwts., thereafter 10s. for every 5 cwts. or part thereof. For example, the extra for plates of 6 tons 3 cwts. would be £2, 10s.

All plates which are not rectangular will be considered sketches. Of these 15 per cent. will be allowed free, any in excess of 15 per cent. being charged . . . . . 25s. per ton.

Over $1\frac{1}{2}$ inches to $1\frac{3}{4}$ inches thick inclusive . . 10s. ,,
,, $1\frac{3}{4}$ inches to $1\frac{3}{4}$ inches ,, . . . 20s. ,,
,, $1\frac{3}{4}$ inches to 2 inches ,, . . . 30s. ,,

For the undermentioned minimum tensile strains:—

A minimum tensile strain of not more than			$28\frac{1}{2}$	tons,	5s. ,,
,,	,,	,,	29	,,	10s. ,,
,,	,,	,,	30	,,	20s. ,,
,,	,,	,,	31	,,	40s. ,,
,,	,,	,,	32	,,	60s. ,,
,,	,,	,,	33	,,	80s. ,,
,,	,,	,,	34	,,	100s. ,,
,,	,,	,,	35	,,	120s. ,,

For the undermentioned low tensile strains—

Plates specified not to exceed	27	tons per square inch,	10s.	,,
,, ,,	26	,, ,,	20s.	,,
,, ,,	25	,, ,,	40s.	,,

A minimum elongation, 25 per cent. on 10-inch	20s.	per ton.
,, ,, 25 ,, 8-inch	15s.	,,
Plates 20 feet long or over, and at the same time 6 feet wide or over, or plates 50 cwts. each or over, to pass Board of Trade Survey, or equal thereto	10s.	,,
Admiralty Survey, or equal thereto	20s.	,,
When the phosphorus and sulphur limit is ·035	70s.	,,
In every case where tensile limit narrowed to 2 tons	20s.	,,
In every case where the range of tensile is over 2 and not exceeding 3 tons	10s.	,,

Testing fees, over and above Lloyd's fees, or where they exceed 1s. per ton, will be charged extra in full.

Extras on Bars.

ROUNDS AND SQUARES.

Rivet steel bars, $\frac{3}{8}$ inch to be taken at basis price.

Under $\frac{5}{8}$ inch to $\frac{1}{2}$ inch	5s.	per ton.
,, $\frac{1}{2}$ inch to $\frac{7}{16}$ inch	10s.	,,
,, $\frac{7}{16}$ inch to $\frac{3}{8}$ inch	20s.	,,
Over 3 inches to $3\frac{3}{4}$ inches	5s.	,,

$3\frac{7}{8}$ inches and up, subject to arrangement.

Stay bars to pass Board of Trade or Admiralty Survey	20s.	,,
,, ,, Lloyd's or Bureau Veritas Survey	10s.	,,
Turning Quality	20s.	,,
Double Survey	1s.	,, extra

Ordinary cut lengths 5 feet and over 3 feet, 5s. per ton extra. 3 feet to 2 feet, 10s. per ton extra. Special extras will be charged for cutting to short lengths, or to exact length, and for special straightening.

Extras for testing fees and for restricted tensile limits, the same for bars as for plates.

Plain Rolled Steel Shafts.

(As supplied by J. Spencer & Son, Newburn.)

10 inches to $10\frac{1}{2}$ inches	30 to 35 feet long.	
11 ,, $11\frac{1}{2}$,,	26 ,, 30	,,
12 ,, $12\frac{1}{2}$,,	22 ,, 25	,,
13 ,, $13\frac{1}{2}$,,	35 ,, 40	,,
14 ,, $14\frac{1}{2}$,,	30	,,
15 ,, $15\frac{1}{2}$,,	25 ,, 30	,,
16 ,, $16\frac{1}{2}$,,	20 ,, 25	,,

These shafts are now supplied with flanges formed by upsetting, etc., by hydraulic presses.

Table CXXXV.—Limits of Safe Working Stresses on Various Metals (Tensile Continuous Service).

Metal or Alloy.	Nature of the Load.	At revolutions per minute, not exceeding									
		60	100	200	300	400	500	600	700	800	1000
Iron castings of good quality,	Steady	5,150	5,000	4,800	4,700	4,600	4,550	4,500	4,460	4,450	4,440
	Intermittent	4,700	4,500	4,200	3,940	3,750	3,600	3,440	3,300	3,200	3,000
	Alternating	3,900	3,700	3,350	3,100	2,900	2,700	2,550	2,400	2,300	2,200
Mild steel castings, good quality,	Steady	15,200	14,650	13,800	13,100	12,550	12,050	11,600	11,200	10,800	10,150
	Intermittent	13,450	12,800	11,950	11,250	10,650	10,150	9,650	9,200	8,850	8,200
	Alternating	11,150	10,550	9,750	8,950	8,400	7,800	7,330	6,900	6,500	6,000
Mild steel bars and plates,	Steady	16,800	16,500	15,900	15,450	15,100	14,800	14,550	14,350	14,150	13,800
	Intermittent	14,350	14,000	13,400	12,900	12,500	12,200	11,930	11,700	11,500	11,150
	Alternating	12,100	11,750	11,100	10,600	10,250	9,900	9,600	9,360	9,150	8,800
High tensile steel bars and forgings, nickel, chrome, vanadium,	Steady	26,800	26,350	25,500	24,800	24,260	23,800	23,350	22,950	22,550	21,750
	Intermittent	24,350	23,850	22,950	22,250	21,700	21,200	20,750	20,300	19,900	19,100
	Alternating	22,100	21,650	20,700	20,000	19,420	18,900	18,450	18,050	17,700	16,950
Bronze, or gun-metal, Admiralty alloy,	Steady	7,800	7,600	7,200	6,930	6,700	6,500	6,350	6,200	6,100	5,950
	Intermittent	7,000	6,800	6,350	6,000	5,660	5,400	5,150	4,930	4,720	4,350
	Alternating	5,900	5,600	5,100	4,650	4,300	4,000	3,750	3,500	3,300	3,000
High tensile bronzes, carefully cast,	Steady	12,700	12,500	12,100	11,700	11,550	11,250	11,050	10,850	10,700	10,400
	Intermittent	11,450	11,000	10,300	9,800	9,350	9,000	8,700	8,400	8,150	7,500
	Alternating	9,500	9,000	8,300	7,750	7,300	6,930	6,600	6,300	6,000	5,450
High tensile bronze bars, &c., unannealed,	Steady	22,500	21,600	20,800	20,100	19,600	19,200	18,900	18,700	18,500	18,140
	Intermittent	19,500	19,000	18,000	17,200	16,600	16,030	15,500	15,000	14,500	13,600
	Alternating	15,500	15,000	14,000	13,200	12,560	12,000	11,500	11,000	11,600	10,750
High tensile bronze bars, &c., annealed,	Steady	15,250	15,000	14,520	14,040	13,860	13,500	13,250	13,020	12,860	12,500
	Intermittent	13,750	13,200	12,360	11,760	11,120	10,800	10,440	10,089	9,800	9,100
	Alternating	11,400	10,900	9,960	9,300	8,760	8,320	7,920	7,560	7,250	6,750

For shearing stresses, multiply the above by 1·62 for cast iron; 0·8 for steel; strong zinc bronzes, 0·60; delta metal, 0·7; phosphor bronze, cast, 1·10; rolled, 0·63.

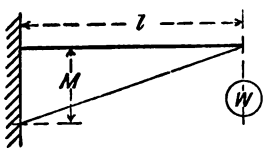
BENDING MOMENTS, Etc.

Table CXXXVI.—Graphic Representation of Bending Moments and Shearing Forces in Simple Cases.

Bending Moment.	Shearing Force.

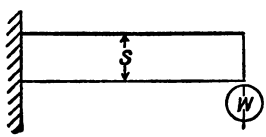
f = greatest permissible stress on material; z = modulus of section —*see* Table CI.; M = greatest bending moment; S = greatest shearing force.

Beams fixed at one end.

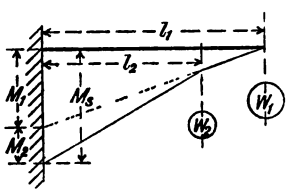


$M = Wl$

Working load $= \frac{fz}{l}$

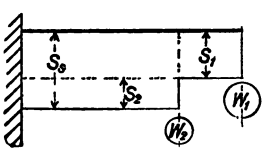


$S = W$



$M_1 = W_1 l_1$; and $M_2 = W_2 l_2$

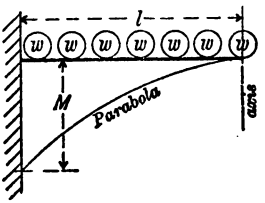
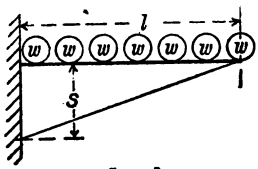
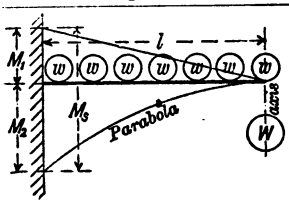
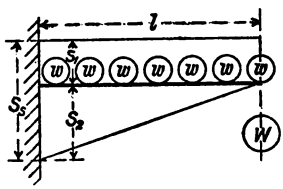
$M_3 = W_1 l_1 + W_2 l_2$



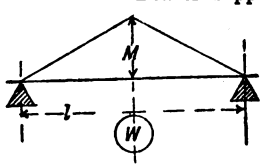
$S_1 = W_1$; and $S_2 = W_2$

$S_3 = W_1 + W_2$

Beams fixed at one end—*continued.*

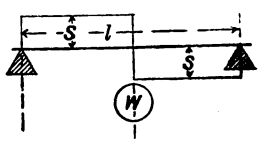
Bending Moment.	Shearing Force.
	
$M = \frac{wl^2}{2}$	$S = wl$
Working load (per unit of length) $= \frac{2f_s}{l^2}$	
	
$M_1 = Wl$ $M_2 = \frac{wl}{2}$ $M_3 = Wl + \frac{wl}{2}$	$S_1 = W$; and $S_2 = wl$ $S_3 = W + wl$

Beams supported at both ends.



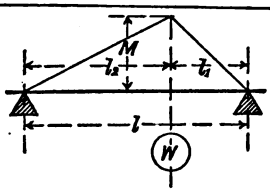
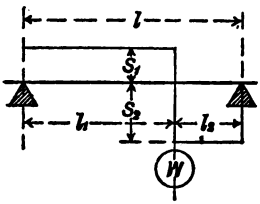
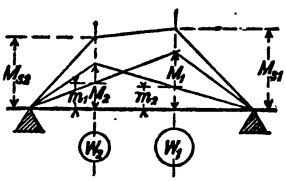
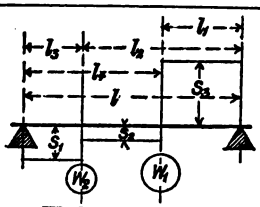
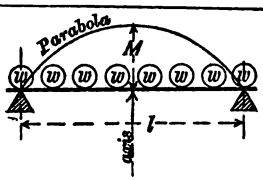
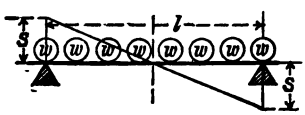
$$M = W\frac{l}{4}$$

Working load $= 4\frac{f_s}{l}$

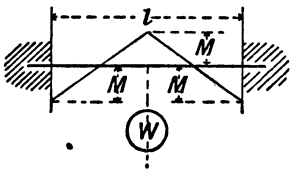
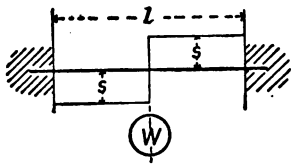
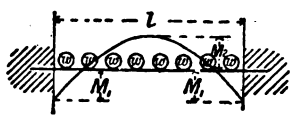
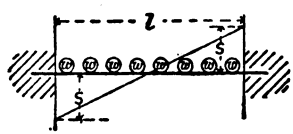


$$S = \frac{W}{2}$$

Beams supported at both ends—*continued.*

Bending Moment.	Shearing Force.
	
$M = W\frac{l_1 l_2}{l}$	$S_1 = W\frac{l_2}{l}$; $S_2 = W\frac{l_1}{l}$
Working load $= \frac{fzl}{l_1 l_2}$	
	
$M_{s1} = M_1 + m_2$ $M_{s2} = M_2 + m_1$	$S_1 = \frac{W_1 l_1 + W_2 l_2}{l}$ $S_2 = \frac{W_1 l_1 + W_2 l_2}{l} - W_2$ $S_3 = \frac{W_1 l_1 + W_2 l_2}{l} - (W_1 + W_2)$ $= -\frac{W_1 l_4 + W_2 l_2}{l}$
	
$M = \frac{wl^2}{8}$	$S = \frac{wl}{2}$
Working load (per unit of length) $= \frac{8fz}{l^2}$	

Beams fixed at both ends.

Bending Moment.	Shearing Force.
 In this case the bending moments at M, M and M are equal, and have the value $M=\frac{Wl}{8}$. Working load $=\frac{8f^s}{l}$.	 $S=\frac{W}{2}$.
 Bending moments at ends (M_1) $M_1=\frac{wl^2}{12}$. Bending moment at centre (M_2) $M_2=\frac{wl^2}{24}$. Working load (per unit of length) $=\frac{12f^s}{l^2}$.	 $S=\frac{wl}{2}$.

Deflection of Beams of Uniform Section.

$D=\frac{1}{8}\left(\frac{WL^3}{EI}\right)$ for a beam fixed at one end and loaded at the other.

$D=\frac{1}{8}\left(\frac{WL^3}{EI}\right)$ " " " " uniformly.

$D=\frac{1}{192}\left(\frac{WL^3}{EI}\right)$ " fixed at both ends and loaded in the middle.

$D=\frac{1}{384}\left(\frac{WL^3}{EI}\right)$ " " " " uniformly.

$D=\frac{1}{48}\left(\frac{WL^3}{EI}\right)$ " supported at both ends and loaded in the middle.

$D=\frac{5}{384}\left(\frac{WL^3}{EI}\right)$ " " " " uniformly.

Where D is deflection in inches; W the total load in lbs.; L the length in inches; E the modulus of elasticity; and I the moment of inertia of section.

Average value of E is as follows:—For cast-iron, 17,000,000; wrought-iron, 27,000,000; wrought-steel, 29,000,000.

For values of I see page 467, 468.

Table CXXXVII.—Moment of Inertia, Modulus, &c., of some Sections.

The plane of bending is supposed perpendicular to plane of paper, and parallel to side of page.

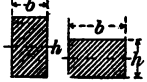
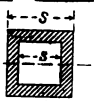
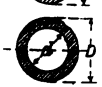
Form of section.	Area of section. A	Moment of inertia of section about axis through centre of gravity. I	Square of rads. of gyration of section. $\rho^2 = \frac{I}{A}$	"Modulus" of section. $Z = \frac{I}{y}$
	s^2	$\frac{s^4}{12}$	$\frac{s^2}{12}$	$\frac{s^3}{6}$
	bh	$\frac{bh^3}{12}$	$\frac{h^3}{12}$	$\frac{bh^2}{6}$
	$S^2 - s^2$	$\frac{S^4 - s^4}{12}$	$\frac{S^2 + s^2}{12}$	$\frac{1}{6}\left(\frac{S^4 - s^4}{S}\right)$
	$b(H - h)$	$\frac{b(H^3 - h^3)}{12}$	$\frac{H^2 + hH + h^2}{12}$	$\frac{b(H^3 - h^3)}{6H}$
	$\cdot 7854 d^2$	$\cdot 0491 d^4$	$\frac{d^2}{16}$	$\cdot 0982 d^3$
	$\cdot 7854 (D^2 - d^2)$	$\cdot 0491 (D^4 - d^4)$	$\frac{D^2 + d^2}{16}$	$\cdot 0982\left(\frac{D^4 - d^4}{D}\right)$

Table CXXXVII.—Moment of Inertia, Modulus, &c., of some Sections—*continued.*

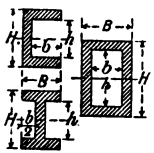
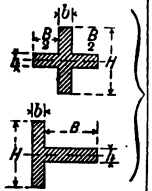
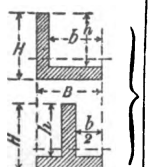
Form of section.	Area of section. A	Moment of inertia of section about axis through centre of gravity. I	Square of rads. of gyration of section. $\rho^2 = \frac{I}{A}$	"Modulus" of section. $Z = \frac{I}{y}$
	$BH - bh$	$\frac{BH^3 - bh^3}{12}$	$\frac{1}{12}\left(\frac{BH^3 - bh^3}{BH - bh}\right)$	$\frac{BH^3 - bh^3}{6H}$
	$Bh + bH$	$\frac{Bh^3 + bH^3}{12}$	—	$\frac{Bh^3 + bH^3}{6H}$
	$BH - bh$	$\frac{(BH^2 - bh^2)^2 - 4BHbh(H-h)^2}{12(BH - bh)}$	—	$\frac{(BH^2 - bh^2)^2 - 4BHbh(H-h)^2}{6(BH^2 - bh^2)}$

Table CXXXVIII.—Forms of Beams of Uniform Strength.

Breadth (b) uniform throughout.

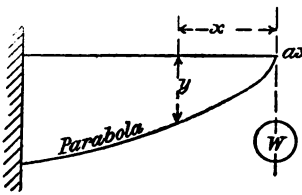
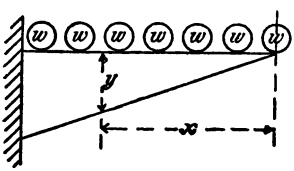
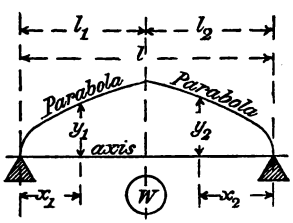
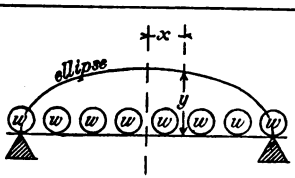
Elevation of arrangement.	Equation for dimensions.
	$y = \sqrt{\frac{6Wx}{bf}}$
	$y = \sqrt{\frac{3wx^2}{bf}}$
	$y_1 = \sqrt{\frac{6Wl_2x_1}{bfl}}$ $y_2 = \sqrt{\frac{6Wl_1x_2}{bfl}}$
	$y = \sqrt{\frac{3w(l^2-4x^2)}{4bf}}$

Table CXXXVIII.—Forms of Beams of Uniform Strength—*continued.*

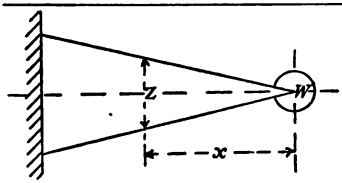
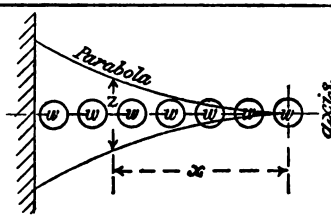
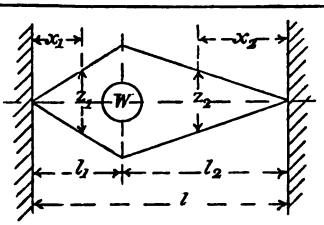
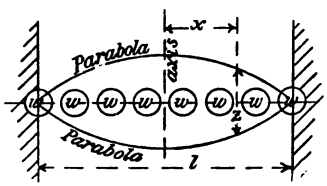
Depth (h) uniform throughout.	
Plan of arrangement.	**Equation for dimensions.**
	$z = \frac{6Wx}{h^2f}$
	$z = \frac{3wx^2}{h^2f}$
	$z_1 = \frac{6Wl_2x_1}{h^2fl}$ $z_2 = \frac{6Wl_1x_2}{h^2fl}$
	$z = \frac{3w(l^2 - 4x^2)}{4h^2f}$

Table CXXXIX.—The Greatest Fluid Test Pressure that should be put on Flat Surfaces of Good Cast Iron in lbs. per sq. inch.

Thickness, Inch.	Breadth of Strip in Inches.																											
	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30
0·3	304	135	76	48	34	25	19	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...
0·4	540	240	135	86	60	44	34	27	22	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...
0·5	944	375	238	135	93	69	53	42	34	28	23	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...
0·6	1215	540	304	194	135	99	76	60	49	40	34	29	25	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...
0·7	1654	735	413	265	184	135	103	82	66	55	49	40	34	30	26	...	...	...	...	...	...	...	...	...	...	...	...	...	...
0·8	2160	960	540	346	240	176	135	107	86	71	60	51	44	38	34	30	26	...	...	...	...	...	...	...	...	...	...	...	...
0·9	2734	1215	683	437	304	223	171	135	109	90	76	64	56	48	42	37	34	31	27	...	...	...	...	...	...	...	...	...	...
1·0	3375	1500	844	540	372	276	211	166	135	112	94	80	69	60	52	47	42	38	34	30	27	...	...	...	...	...	...	...	...
1·1	4082	1815	1020	653	454	333	255	202	163	135	114	99	83	72	64	56	50	45	41	37	34	31	28	...	...	...	...	...	...
1·2	...	2160	1215	778	540	397	304	240	194	161	135	115	99	86	76	67	60	54	49	44	40	37	34	31	...	...	...	...	...
1·3	...	2534	1425	912	633	465	356	282	228	188	158	135	116	101	89	79	70	63	57	51	47	43	40	37	34	...	...	...	...
1·4	...	2940	1654	1058	702	540	414	327	265	219	184	157	135	118	103	91	81	73	66	60	55	50	46	42	39	36	34	...	...
1·5	...	3374	1899	1215	833	620	475	374	304	251	211	180	155	135	119	105	94	84	76	69	63	57	52	48	43	40	38	36	34
1·6	...	3840	2160	1382	960	705	540	427	346	286	240	204	177	153	135	120	107	95	86	78	71	65	60	55	51	47	44	41	38
1·7	...	4335	2438	1561	1084	796	609	482	390	318	271	231	200	174	153	135	120	108	97	89	81	74	68	62	58	54	50	46	43
1·8	...	...	2734	1750	1215	893	683	540	437	361	304	260	223	195	171	151	135	121	109	99	90	83	76	70	65	60	56	52	49
1·9	...	...	3046	1949	1354	996	761	602	487	403	330	288	250	216	190	169	150	135	122	110	100	92	84	78	72	67	62	58	54
2·0	...	...	3374	2160	1500	1102	844	667	540	446	357	320	276	240	211	187	166	150	135	122	112	102	94	86	80	74	69	64	60

If of high-class metal (9 tons tensile), multiply by 1·4.

For a Steady Load the maximum Working Pressure should not exceed 60 per cent. of the above; for an Intermittent Load, 46 per cent.; and for an Alternating, 35 per cent. If made of the best iron and twice melted, the pressures may be 33 per cent. higher.

Table CXL.—The Greatest Fluid Test Pressure on Flat Surfaces of Rolled Mild Steel or Strong Bronzes when unsupported.

Thickness in 32nds of an inch.	Breadth of Strips in Inches.															
	4.	5.	6.	7.	8.	9.	10.	11.	12.	13.	14.	15.	16.	17.	18.	19.
3	28	18	...	...	...	...	...	...	...	...	...	...	...	...	...	...
4	50	32	22	...	...	...	...	...	...	...	...	...	...	...	...	...
5	78	50	35	25	...	...	...	...	...	...	...	...	...	...	...	...
6	112	72	50	37	28	...	...	...	...	...	...	...	...	..	...	...
7	153	98	68	50	38	30	...	...	...	...	...	...	...	...	...	...
8	200	128	89	65	50	40	32	...	...	...	...	...	...	...	...	...
9	253	162	112	83	63	50	40	34	...	...	...	...	...	...	...	...
10	312	200	139	102	78	62	50	41	35	...	...	...	...	...	...	...
11	378	242	168	123	94	75	60	50	42	36	...	...	...	...	...	...
12	450	288	200	147	112	89	72	60	50	42	37	...	...	...	...	...
13	528	338	235	172	132	104	85	70	59	50	43	38	...	...	...	...
14	612	392	271	200	153	121	98	81	68	58	50	43	38	...	...	...
15	...	450	312	238	176	139	113	93	78	67	57	50	44	39	...	...
16	...	512	355	261	200	158	128	106	89	76	65	57	50	44	40	...
17	...	578	401	295	226	179	145	119	100	85	74	65	56	50	45	40
18	...	...	450	330	253	200	162	134	112	96	83	72	63	56	50	45
19	...	...	501	368	282	223	181	149	125	107	92	80	70	63	56	50
20	...	...	555	409	312	247	200	166	139	118	102	89	78	70	62	55
21	...	...	...	450	344	272	221	182	153	130	112	98	86	76	68	61
22	...	...	...	493	378	298	242	200	168	143	123	107	94	83	74	67
23	...	...	...	540	413	327	265	219	184	156	135	117	103	91	81	73
24	...	...	...	...	450	356	288	238	200	170	147	128	112	100	89	79

N.B.—When secured at the edges or stiffened by flanges or continuous, multiply by 2. The Working Pressure with a Steady Load may be 60 per cent. of the test; for an Intermittent Load, 46 per cent.; and for Alternating, 35 per cent.

Table CXLI.—The Greatest Fluid Test Pressure that should be put on Flat Surfaces of Phosphor Bronze Castings in lbs. per sq. inch.

Thickness, Inch.	Breadth of Strip in Inches.																										
	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30
0·15	140	62	35	22	16	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...
0·20	250	111	62	40	28	20	16	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...
0·25	390	174	97	62	43	32	24	20	16	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...
0·30	563	250	141	90	63	46	35	28	23	19	16	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...
0·35	781	347	195	125	87	63	49	39	31	24	20	18	16	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...
0·40	1000	444	250	160	111	82	62	49	40	33	28	24	20	18	16	...	...	...	...	...	...	...	...	...	...	...	...	...	...
0·45	1262	561	315	202	140	103	79	62	50	42	34	30	26	23	20	18	16	...	...	...	...	...	...	...	...	...	...	...	...
0·50	1562	694	390	250	173	128	98	77	63	52	43	37	32	28	24	21	19	17	16	...	...	...	...	...	...	...	...	...	...
0·55	1890	840	473	302	210	154	118	93	75	62	52	45	38	34	30	26	23	21	19	17	16	...	...	...	...	...	...	...	...
0·60	2250	1000	562	360	250	183	140	111	90	73	62	53	46	40	35	31	28	25	23	21	19	17	16	...	...	...	...	...	...
0·65	2640	1174	660	422	294	215	165	130	106	87	73	62	54	47	41	36	32	29	26	24	22	20	19	18	16	...	...	...	...
0·70	3056	1358	764	489	339	250	191	151	122	101	85	72	62	54	48	42	38	34	30	27	25	23	21	20	19	18	16	...	...
0·75	3512	1562	878	562	390	287	219	173	141	116	98	83	72	62	55	49	44	39	35	32	29	26	24	22	20	19	18	17	16
0·80	4000	1778	1000	640	444	327	250	198	160	132	111	95	82	71	62	55	49	44	40	36	33	30	28	26	24	22	20	19	18
0·85	...	2007	1129	722	502	379	282	223	181	149	125	107	92	80	70	62	56	50	45	41	37	34	31	29	27	25	23	21	20
0·90	...	2250	1266	810	563	413	316	250	203	167	141	120	103	90	79	70	62	56	51	46	42	38	35	32	30	28	26	24	22
0·95	...	2507	1410	903	627	460	352	277	226	187	157	133	115	100	88	78	70	63	56	51	47	43	39	36	33	31	29	27	25
1·00	...	2778	1562	1000	700	511	391	309	250	207	174	150	122	111	99	86	77	69	63	5 9	52	48	44	40	37	34	32	30	28
1·10	...	3361	1891	1210	840	617	473	373	303	250	210	180	155	134	118	105	93	84	76	69	63	59	53	49	46	42	39	36	34
1·20	...	4000	2250	1440	1000	735	563	444	360	298	250	213	184	160	140	125	111	100	90	82	75	68	63	58	54	50	46	43	40

For a Steady Load the maximum Working Pressure should not exceed 60 per cent. of the above; for an Intermittent Load, 46 per cent.; and for an Alternating, 33 per cent. Admiralty Bronze castings should not be tested beyond 75 per cent. of the above.

EXPANSION OF METALS BY HEAT.

The following table shows approximately the extent to which metals, &c., expand under the action of heat:—

Table CXLII.—Expansion of Metals, &c., for rise of Temperature of 180° F.

Material.	Change of length.	
	Fraction of total length.	Inches per foot. (C)
Cast-iron, ...	·00117	·0140
Wrought-iron, ...	·00122	·0146
Steel, ...	·00120	·0146
Copper, ...	·00182	·0218
Gun-metal, ...	·00187	·0224
Fire-brick, ...	·000428	·0050

If t_1 and t_2 be the highest and lowest temperatures to which the object is exposed, the *alteration* in length is given by,—

$$\text{Alteration in length (increase or diminution)} = C \times \left(\frac{t_1 - t_2}{180}\right) \times L$$

where L is length in feet, value of C is taken from above table. and alteration of length is given in inches.

The expansion of metals, per degree rise of temperature, increases slightly as higher temperatures are reached, but for all practical purposes it may be assumed to be constant.

Effect of temperature on strengths of metals.—At 400° F. almost all alloys of copper, tin, and zinc lose from 15 to 20 per cent. of their ultimate strength ; but beyond this temperature some of the copper-zinc and copper-aluminium alloys weaken very rapidly.

Copper is similarly affected, and loses 8 to 10 per cent. of its strength at 400° F.

Cast-iron is practically unaffected up to 400° F.

Wrought-iron and mild steel *gain* from 10 to 15 per cent. in ultimate strength at 400° F.

Up to 400° F. the changes seem to progress regularly with the changes of temperature.

Table CXLIII.—Effect of Temperature on Metals.

Metal or Alloy.	At 100° F.		At 250° F.		At 350° F.		At 400° F.		At 450° F.		Melting point. F.
	Ultimate strength.	Extension.	Ultimate strength.	Extension.	Ultimate strength.	Extension.	Ultimate strength.	Extension.	Ultimate strength.	Extension.	
		Per cent.		Per cent.		Per cent.		Per cent.		Per cent.	
Copper (0·2 per cent. arsenic),	30,325	30·3	31,000	26·2	30,950	21·1	29,750	27·0	29,000	29·0	2,000°
Admiralty bronze, cast,	32,000	12·0	30,800	9·3	29,800	8·25	15,700	0·75	15,500	0·2	1,700°
Mercantile bronze (soft), cast,	29,000	26·0	28,800	25·8	28,000	23·0	27,750	25·0	9,700	1·2	1,680°
Naval brass, rolled,	58,700	18·8	54,500	17·6	52,600	16·6	52,000	16·0	52,000	14·0	1,640°
Phosphor bronze, wheel brand, cast,	39,000	17·8	37,000	15·0	29,400	7·0	27,000	5·0	27,000	5·0	1,900°
Phosphor bronze, wheel brand, rolled,	68,250	6·8	68,000	7·0	67,750	7·5	64,500	8·1	60,000	9·0	1,900°
Parsons'immadium, cast,	89,600	20·0	...	...	...	...	56,400	25·0	51,500	22·5	1,660°
Aich's metal, rolled,	55,250	...	47,000	...	42,300	...	39,750	...	37,000	...	1,650°
Bull's metal, ,,	70,000	16·0	63,600	13·6	58,700	13·0	56,750	14·6	...	...	1,800°
Aluminium, ,,	25,300	...	20,250	...	15,500	...	13,300	...	11,500	...	*1,160°
Melloid, ,,	98,000	12·0	...	...	...	...	90,700	15·0	87,300	15·0	...

* Plastic State.

Note.—Copper at 500° has an ultimate tensile of 10·8 tons, with 40 per cent. extension.
Immadium at 500° ,, ,, 23·5 ,, 22·5 ,, ,,
Copper at 650° ,, ,, 9·7 ,, 33·5 ,, ,,

MELTING POINTS OF METALS.

Table CXLIV. gives the melting points of the more ordinary metals as determined by the latest researches.

Table CXLIV.—Melting Points of Metals, etc.

Materials.	Degrees.		Materials.	Degrees.	
	Centi.	Fahr.		Centi.	Fahr.
Water . .	0	32	Pig iron (white) .	1135	2075
Mercury . .	−39	−39	,, (grey) .	1121	2050
Paraffin Wax .	54	129	,, (forge) .	1220	2192
Sulphur . .	115	239	Grey iron, 2nd cast	1240	2264
Tin . . .	235	455	Steel, carbon 2%.	1315	2400
Bismuth . .	270	518	,, ,, low .	1580	2876
Lead . . .	330	630	Wrought iron welds	1300	2550
Zinc . . .	415	779	,, ,, melts	1600	2910
Antimony . .	621	1150	Pure iron ,,	1525	2769
Aluminium .	625	1157	Nickel ,,	1420	2588
Brass. . .	900	1650	Vanadium ,,	1730	3145
Bronze or gun-metal . .	905	1660	Platinum ,,	1775	3217
			Heat, dull red	700	1290
Bronze phosphor	1038	1900	,, cherry-red .	800	1470
Silver . .	954	1750	,, orange .	1140	2100
Copper . .	1054	1929	,, white .	1320	2370
Cobalt . .	1467	2672	,, dazzling .	1500	2730

Table CXLIVa.—Some Alloys which Melt at Low Temperatures.

Designation.	Composition.				Melts at	
	Bismuth.	Lead.	Tin.	Cadmium.	C.°	F.°
B.C.L.T. No. 1 .	50·0	25·0	12·5	12·5	65	149
,, ,, 2 .	50·1	26·6	13·3	10·0	70	158
,, ,, 3 .	27·5	27·5	10·0	34·5	75	167
,, ,, 4 .	...	25·0	50 0	25·0	86	187
B.L.T. No. 1 .	50·0	31·2	18·7	...	94	201
,, ,, 2 .	40·0	40·0	20·0	...	113	235
,, ,, 3 .	30·8	38·4	30·8	...	130	266
,, ,, 4 .	25·0	50·0	25·0	...	149	300
,, ,, 5 .	16·0	36·0	48·0	...	155	311
,, ,, 6 .	13·3	46·6	40·1	...	165	329
,, ,, 7 .	12·5	50·0	37·5	...	178	352

Table CXLIVb.—Specific Heat of various Metals, &c., Water being Unity.

Material.	Specific Heat.	Material.	Specific Heat.
Aluminium,	0·234	Oxygen (equal weight),	0·156
Antimony,	0·051	Steam (saturated) (equal weight),	0·305
Bismuth,	0·031	Steam (superheated) (equal weight),	0·370
Brass (condenser tubes, etc.),	0·094	Alcohol,	0·659
Copper,	0·095	Ether,	0·521
Gold,	0·032	Mercury,	0·033
Iron (castings),	0·130	Olive oil,	0·310
Iron and steel, wrought,	0·110	Petroleum,	0·434
Lead,	0·031	Petrol,	0·303
Manganese,	0·144	Turpentine,	0·416
Nickel,	0·109	Water at 212° F. (=15 lbs. absolute pressure),	1·0130
Platinum,	0·033	Water at 307° F. (=75 lbs. absolute pressure),	1·0270
Silver,	0·057	Water at 328° F. (=100 lbs. absolute pressure),	1·0308
Steel (high carbon),	0·117	Water at 344° F. (=125 lbs. absolute pressure),	1·0338
Tin,	0·057	Water at 358° F. (=150 lbs. absolute pressure),	1·0368
Tungsten,	0·036	Water at 371° F. (=175 lbs. absolute pressure),	1·0393
Zinc,	0·096	Water at 381° F. (=200 lbs. absolute pressure),	1·0415
Brick work,	0·192	Water at 391° F. (=225 lbs. absolute pressure),	1·0436
Coal (Welsh),	0·201	Water at 401° F. (=250 lbs. absolute pressure),	1·0460
,, Bituminous,	0·241	Water at 409° F. (=275 lbs. absolute pressure),	1·0480
Coke,	0·203	Water at 417° F. (=300 lbs. absolute pressure),	1·0500
Glass,	0·198		
Graphite,	0·202		
Marble and limestone,	0·217		
Air at constant pressure (equal weight),	0·238		
Air at constant volume (equal weight),	0·169		
Carbonic acid (Co_2) (equal weight),	0·171		
Carburetted hydrogen (equal weight),	0·468		
Nitrogen (equal weight),	0·174		

Table CXLIVc.—Thermal Conductivity of Metals at 18° to 20° C.

Silver	100·0	Cobalt	14·7
Copper	91·8	Nickel	14·0
Gold	53·2	Iron	11·9
Aluminium	31·3	Lead	8·5
Zinc	28·1	Platinum	8·4
Tin	15·2	Bismuth	1·8

Table CXLIVd.—Electrical Resistance.

Materials.	Resistance in Microhms at 0° C. per cubic inch.
Silver, annealed	0·5874
Copper, ,,	0·6378
,, hard drawn	0·6382
Gold, ,,	0·8177
Aluminium, annealed	1·1370
,, hard drawn	...
Zinc, ,,	2·1970
Cobalt	3·189
Platinum, annealed	3·5360
Iron ,,	3·7940
Nickel ,,	4·8660
Tin, pressed	5·1570
Lead ,,	7·6650
Antimony, ,,	13·8600
Bismuth, ,,	51·2200
Mercury, liquid	37·1400
German silver	11·8000
Manganese copper (70Cu + 30Mn)	40·5000
Nickel manganese copper (73Cu + 24Mn + 3Ni)	18·8000

Note.—The recording resistance pyrometer is now so perfect that differences of one-fifth of 1° F. in the neighbourhood of, say, 2000° F. can be measured. But the results obtained as to the melting points of metals by different observers still differ by some 30° to 50° F., partly owing to slight differences in the conditions, and partly owing to the fact that no metal solidifies entirely at one temperature. In the case of pure metals the "freezing range" is small, but in that of alloys it is often considerable, and the "freezing curve" shows several distinct freezing points.

In the case of bronze containing 90 per cent. copper and 10 per cent. in, for instance, there are two well-marked freezing points,—one at bout 1800° F. and another at about 1470° F.,—the "freezing range" eing thus about 330° F.

For further information see Reports of Alloys Research Committee of Inst. of Mech. Engineers.

WEIGHTS OF MATERIALS.

Table CXLV.—Weights of Materials (Summary).

Material.	Weight of a cubic foot in lbs.	Weight of a cubic inch in lbs.
Air (32° F. and 14·7 lbs. pressure)	·0807	...
Water, pure,	62·4	·036
Water, river,	63	...
Water, sea,	64	·037
Colza, linseed, or olive oil,	57–58	...
Mineral oils,	56	...
Tallow,	58	...
Waste (moderately pressed),	11	...
Elm, pine, or fir timber,*	30–40	...
Beech, ash, or birch, ,,	40–45	...
Oak or teak, ,,	45–55	...
Greenheart, ,,	about 65	...
Lignum-vitæ, ,,	about 80	...
Fire brick,	about 140	...
Wrought-iron bars or plates,	485	·281
Staffordshire plates,	480	·277
Iron forgings (large),	477	·276
Steel bars and plates,	490	·284
Steel forgings (large),	487	·282
Steel—Whitworth compressed,	496	·287
Cast-iron,	450	·260
Cast-steel (mild),	490	·284
Sheet copper,	550	·318
Gun-metal,	545	·315
Muntz-metal,	512	·296
Naval brass (rolled),	530	·307
White metal (Babbit's),	456	·263
Zinc cast,	435	·252
Zinc rolled,	450	·260
Lead cast,	708	·408
Lead sheet,	711	·411

A plate of cast-iron	1 foot square	and	1 inch thick	weighs 37·5 lbs.
" wrought-iron	"		"	40 "
" cast-steel (mild)	"		"	41 "
" wrought-steel (mild)	"		"	41 "
" gun-metal	"		"	45 "
" rolled brass	"		"	44 "

The Admiralty reckon 40 cubic feet of bunker space as 1 ton, but the usual allowance is 45 cubic feet to the ton; the actual average bulk of a ton is about 43 cubic feet, but if taken at 45 cubic feet a fair allowance is made for the upper portions under the deck-beams, which cannot be filled.

* The following are the exact weights of the various kinds of pine wood in general use, as derived from a series of careful experiments made by Mr Seaton :—

Yellow pine,	very dry,	24	lbs. per cubic foot.
"	in planks, seasoned,	28	" "
Baltic red pine,	" "	30·5	" "
Oregon pine,	" "	35	" "
Kauri pine,	" "	42	" "
Pitch pine,	" "	45	" "

Table CXLVI.—Table of the Weight of Round and Square Wrought-Iron Bars in lbs. per lineal foot.

Dia. or Side.	Weight in lbs.		Dia. or Side.	Weight in lbs.		Dia. or Side.	Weight in lbs.	
	Round.	Square.		Round.	Square.		Round.	Square.
1/16	·093	·117	3 3/4	36·812	46·875	8	167·53	213·33
1/8	·164	·208	7/8	39·306	50·052	1/8	172·81	220·05
3/16	·256	·326				1/4	178·17	226·88
1/4	·368	·469	4	41·884	53·333	3/8	183·61	233·80
5/16	·501	·638	1/8	44·542	56·719	1/2	189·13	240·83
3/8	·654	·833	1/4	47·283	60·208	5/8	194·73	247·97
7/16	·828	1·055	3/8	50·105	63·802	3/4	200·42	255·21
1/2	1·023	1·302	1/2	53·009	67·500	7/8	206·19	262·55
9/16	1·237	1·576	5/8	55·995	71·302			
5/8	1·473	1·875	3/4	59·062	75·208	9	212·04	270·00
11/16	1·728	2·201	7/8	62·212	79·219	1/8	217·97	277·55
3/4	2·004	2·552				1/4	223·98	285·21
13/16	2·300	2·930	5	65·443	83·333	3/8	230·07	292·97
			1/8	68·756	87·552	1/2	236·25	300·83
1	2·618	3·333	1/4	72·151	91·875	5/8	242·51	308·80
1/8	3·313	4·219	3/8	75·628	96·302	3/4	248·85	316·88
1/4	4·090	5·208	1/2	79·186	100·83	7/8	255·27	325·05
3/8	4·949	6·302	5/8	82·827	105·47			
1/2	5·890	7·500	3/4	86·549	110·21	10	261·77	333·33
5/8	6·912	8·802	7/8	90·353	115·05	1/8	268·36	341·72
3/4	8·017	10·208				1/4	275·03	350·21
7/8	9·203	11·719	6	94·238	120·00	3/8	281·77	358·80
			1/8	98·206	125·05	1/2	288·60	367·50
2	10·471	13·333	1/4	102·26	130·21	5/8	295·52	376·30
1/8	11·821	15·052	3/8	106·39	135·47	3/4	302·51	385·21
1/4	13·252	16·875	1/2	110·60	140·83	7/8	309·59	394·22
3/8	14·766	18·802	5/8	114·89	146·30			
1/2	16·361	20·833	3/4	119·27	151·88	11	316·75	403·33
5/8	18·038	22·969	7/8	123·73	157·55	1/8	323·99	412·55
3/4	19·797	25·208				1/4	331·31	421·88
7/8	21·637	27·552	7	128·27	163·33	3/8	338·71	431·30
			1/8	132·89	169·22	1/2	346·20	440·83
3	23·560	30·000	1/4	137·60	175·21	5/8	353·76	450·47
1/8	25·564	32·552	3/8	142·98	181·30	3/4	361·41	460·21
1/4	27·650	35·208	1/2	147·25	187·50	7/8	369·14	470·05
3/8	29·818	37·969	5/8	152·20	193·80			
1/2	32·067	40·833	3/4	157·23	200·21	12	376·95	480·00
5/8	34·399	43·802	7/8	162·34	206·72			

For larger sizes take weight of bar of half the diameter or side and multiply it by four (*vide* page 481).

Table CXLVII.—Weight of Round and Square Bar Steel in lbs. per lineal foot.

Dia. or Side.	Weight in lbs.		Dia. or Side.	Weight in lbs.		Dia. or Side.	Weight in lbs.	
	Round.	Square.		Round.	Square.		Round.	Square.
1/8	·042	·053	3 5/8	35·090	44·678	7 7/8	165·60	210·85
3/16	·094	·120	3/4	37·552	47·813			
1/4	·167	·213	7/8	40·097	51·053	8	170·90	217·60
5/16	·261	·332				1/8	176·29	225·25
3/8	·375	·478	4	42·726	54·400	1/4	181·75	231·41
7/16	·511	·651	1/8	45·438	57·853	3/8	187·30	238·48
1/2	·667	·850	1/4	48·233	61·413	1/2	192·93	245·65
9/16	·845	1·076	3/8	51·112	65·078	5/8	198·65	252·93
5/8	1·043	1·328	1/2	54·075	68·850	3/4	204·45	260·31
11/16	1·262	1·607	5/8	57·121	72·728	7/8	210·33	267·80
3/4	1·502	1·913	3/4	60·250	76·713			
13/16	1·762	2·245	7/8	63·463	80·803	9	216·30	275·40
7/8	2·044	2·603				1/8	222·35	283·10
15/16	2·347	2·988	5	66·759	85·000	1/4	228·48	290·91
			1/8	70·139	89·303	3/8	234·70	298·83
1	2·670	3·400	1/4	73·602	93·713	1/2	241·00	306·85
1/8	3·380	4·303	3/8	77·148	98·229	5/8	248·38	314·98
1/4	4·172	5·313	1/2	80·778	102·85	3/4	253·85	323·21
3/8	5·049	6·428	5/8	84·492	107·58	7/8	260·40	331·55
1/2	6·008	7·650	3/4	88·288	112·41			
5/8	7·051	8·978	7/8	92·169	117·35	10	267·04	340·00
3/4	8·178	10·413				1/8	273·75	348·55
7/8	9·388	11·953	6	96·133	122·40	1/4	280·55	357·21
			1/8	100·18	127·55	3/8	287·44	365·98
2	10·681	13·600	1/4	104·31	132·81	1/2	294·41	374·85
1/8	12·058	15·353	3/8	108·52	138·18	5/8	301·46	383·83
1/4	13·519	17·213	1/2	112·82	143·65	3/4	308·59	392·91
3/8	15·062	19·178	5/8	117·20	149·23	7/8	315·81	402·10
1/2	16·690	21·250	3/4	121·67	154·91			
5/8	18·400	23·428	7/8	126·22	160·70	11	323·11	411·40
3/4	20·195	25·713				1/8	330·50	420·80
7/8	22·072	28·103	7	130·85	166·60	1/4	337·97	430·31
			1/8	135·56	172·60	3/8	345·52	439·93
3	24·033	30·600	1/4	140·36	178·71	1/2	353·15	449·65
1/8	26·078	33·203	3/8	145·24	184·93	5/8	360·87	459·48
1/4	28·206	35·913	1/2	150·21	191·25	3/4	368·68	469·41
3/8	30·417	38·728	5/8	155·26	197·68	7/8	376·56	479·45
1/2	32·712	41·650	3/4	160·39	204·21	12	384·53	489·60

For larger sizes take weight of bar of half the diameter or side and multiply it by four (*vide* page 482).

Table CXLVIIa.—Round Steel Shafts, Weight per foot in lbs.

Diameter.	Weight.	Diameter.	Weight.	Diameter.	Weight.
12¼	400·7	16¼	705·2	20¼	1095
12½	417·2	16½	727·0	20½	1122
12¾	434·1	16¾	749·2	20¾	1150
13	451·3	17	771·7	21	1178
13¼	468·8	17¼	794·6	21¼	1206
13½	486·7	17½	817·8	21½	1234
13¾	504·9	17¾	841·3	21¾	1263
14	523·4	18	865·2	22	1292
14¼	542·2	18¼	889·4	22¼	1322
14½	561·4	18½	913·9	22½	1352
14¾	581·0	18¾	938·8	22¾	1382
15	600·8	19	964·0	23	1413
15¼	621·0	19¼	993·5	23¼	1444
15½	641·6	19½	1015	23½	1475
15¾	662·4	19¾	1042	23¾	1506
16	683·6	20	1070	24	1538

Table CXLVIIb.—Round Hollow Shafts—Bore, Half of Outside Diameter.

Diameter.	Weight.	Diameter.	Weight.	Diameter.	Weight.
ins.		ins.		ins.	
10	200	14½	421	19	723
10½	221	15	450	19½	761
11	242	15½	480	20	802
11½	265	16	513	20½	841
12	288	16½	545	21	883
12½	313	17	579	21½	925
13	338	17½	613	22	969
13½	364	18	649	23	1059
14	391	18½	686	24	1153

Table CXLVIII.—Weight of Flat Wrought-Iron Bars in lbs. per lineal foot.

Breadth of bar (ins.).	Thickness in fractions of an inch.													Breadth of bar (ins.).
	¼	5/16	⅜	7/16	½	9/16	⅝	11/16	¾	13/16	⅞	15/16	1 in.	
1	·83	1·04	1·25	1·46	1·67	1·88	2·08	2·29	2·50	2·71	2·92	3·13	3·33	1
1¼	1·04	1·30	1·56	1·82	2·08	2·34	2·60	2·86	3·13	3·39	3·65	3·91	4·16	1¼
1½	1·25	1·56	1·88	2·19	2·50	2·81	3·13	3·44	3·75	4·06	4·38	4·69	5·00	1½
1¾	1·46	1·82	2·19	2·55	2·92	3·28	3·65	4·01	4·38	4·74	5·10	5·47	5·83	1¾
2	1·67	2·08	2·50	2·92	3·33	3·75	4·17	4·58	5·00	5·42	5·83	6·25	6·67	2
2¼	1·88	2·34	2·81	3·28	3·75	4·22	4·69	5·16	5·63	6·09	6·56	7·03	7·50	2¼
2½	2·08	2·60	3·13	3·65	4·17	4·69	5·21	5·73	6·25	6·77	7·29	7·81	8·33	2½
2¾	2·29	2·86	3·44	4·01	4·58	5·16	5·73	6·30	6·88	7·45	8·02	8·59	9·17	2¾
3	2·50	3·13	3·75	4·38	5·00	5·63	6·25	6·88	7·50	8·13	8·75	9·38	10·00	3
3¼	2·71	3·39	4·06	4·74	5·42	6·09	6·77	7·45	8·13	8·80	9·48	10·16	10·83	3¼
3½	2·92	3·65	4·38	5·10	5·83	6·56	7·29	8·02	8·75	9·48	10·21	10·94	11·67	3½
3¾	3·13	3·91	4·69	5·47	6·25	7·03	7·81	8·59	9·38	10·16	10·94	11·72	12·50	3¾
4	3·33	4·17	5·00	5·83	6·67	7·50	8·33	9·17	10·00	10·83	11·67	12·50	13·33	4
4¼	3·54	4·43	5·31	6·20	7·08	7·97	8·85	9·74	10·63	11·51	12·40	13·28	14·17	4¼
4½	3·75	4·69	5·63	6·56	7·50	8·44	9·38	10·31	11·25	12·19	13·13	14·06	15·00	4½
4¾	3·96	4·95	5·94	6·93	7·92	8·91	9·90	10·89	11·88	12·86	13·85	14·84	15·83	4¾
5	4·17	5·21	6·25	7·29	8·33	9·38	10·42	11·46	12·50	13·54	14·58	15·63	16·67	5

Table CXLIX.—Weight of Flat Rolled Steel Bars in lbs. per lineal foot.

Breadth of bar (ins.).	Thickness in fractions of an inch.													Breadth of bar (ins.).
	¼	5/16	⅜	7/16	½	9/16	⅝	11/16	¾	13/16	⅞	15/16	1 in.	
1	·85	1·06	1·28	1·49	1·70	1·91	2·13	2·34	2·55	2·76	2·98	3·19	3·40	1
1¼	1·06	1·33	1·59	1·86	2·13	2·39	2·66	2·92	3·19	3·45	3·72	3·98	4·25	1¼
1½	1·28	1·59	1·91	2·23	2·55	2·87	3·19	3·51	3·83	4·14	4·46	4·78	5·10	1½
1¾	1·49	1·86	2·23	2·60	2·98	3·35	3·72	4·09	4·46	4·83	5·21	5·58	5·95	1¾
2	1·70	2·13	2·55	2·98	3·40	3·83	4·25	4·68	5·10	5·53	5·95	6·38	6·80	2
2¼	1·91	2·39	2·87	3·35	3·82	4·30	4·78	5·26	5·74	6·22	6·69	7·17	7·65	2¼
2½	2·13	2·66	3·19	3·72	4·25	4·78	5·31	5·84	6·38	6·91	7·44	7·97	8·50	2½
2¾	2·34	2·92	3·51	4·09	4·68	5·26	5·84	6·43	7·01	7·60	8·18	8·77	9·35	2¾
3	2·55	3·19	3·83	4·46	5·10	5·74	6·38	7·01	7·65	8·29	8·93	9·56	10·20	3
3¼	2·76	3·45	4·14	4·83	5·53	6·22	6·91	7·60	8·29	8·98	9·67	10·36	11·05	3¼
3½	2·98	3·72	4·46	5·21	5·95	6·69	7·44	8·18	8·93	9·67	10·41	11·16	11·90	3½
3¾	3·19	3·98	4·78	5·58	6·38	7·17	7·97	8·77	9·56	10·36	11·16	11·95	12·75	3¾
4	3·40	4·25	5·10	5·95	6·80	7·65	8·50	9·35	10·20	11·05	11·90	12·75	13·60	4
4¼	3·61	4·52	5·42	6·32	7·22	8·13	9·03	9·93	10·84	11·74	12·64	13·55	14·45	4¼
4½	3·83	4·78	5·74	6·69	7·65	8·61	9·56	10·52	11·48	12·43	13·39	14·34	15·30	4½
4¾	4·04	5·05	6·06	7·07	8·08	9·08	10·09	11·10	12·11	13·12	14·13	15·14	16·15	4¾
5	4·25	5·31	6·38	7·44	8·50	9·56	10·63	11·69	12·75	13·81	14·88	15·94	17·00	5

Table CXLIXa.—Weight of Rectangular Steel Bars per foot in lbs.

Breadth of bar.	Thickness in inches.												Breadth of bar.
	1¼	1½	1¾	2	2¼	2½	2¾	3	3¼	3½	3¾	4	
6	25·5	30·60	35·7	40·8	45·9	51·0	56·1	61·2	66·3	71·4	76·5	81·6	6
7	29·75	35·70	41·7	47·6	53·6	59·5	65·5	71·4	77·4	83·3	89·3	95·2	7
8	34·00	40·80	47·7	54·4	61·2	68·0	74·8	81·6	88·4	95·2	102·0	108·8	8
9	38·25	45·9	53·6	61·2	68·9	76·5	84·2	91·8	99·5	107·1	114·8	122·4	9
10	42·50	51·0	59·5	68·0	76·5	85·0	93·5	102·0	110·5	119·0	127·5	136·0	10
11	46·75	56·1	65·5	74·8	84·2	93·5	102·9	112·2	121·6	130·9	140·3	149·6	11
12	51·00	61·2	71·4	81·6	91·8	102·0	112·2	122·4	132·6	142·8	153·0	163·2	12
13	55·25	66·3	77·4	88·4	99·5	110·5	121·6	132·6	143·7	154·7	165·8	176·8	13
14	59·50	71·4	83·3	95·2	107·1	119·0	130·9	142·8	154·7	166·6	178·5	190·4	14
15	63·75	76·5	89·3	102·0	114·8	127·5	140·3	153·0	165·8	178·5	191·3	204·0	15
16	68·00	81·6	95·2	108·8	122·4	136·0	149·6	163·2	176·8	190·4	204·0	217·6	16

Table CL.—Weight of Iron Angle-Bars in lbs. per lineal foot.

Sum of flanges (ins.).	Thickness in fractions of an inch.										
	3/16	1/4	5/16	3/8	7/16	1/2	9/16	5/8	11/16	3/4	13/16
2½	1·45	1·88	2·28	2·66	...	...	...	...	...	...	...
2¾	1·60	2·08	2·54	2·97	...	...	...	...	...	...	...
3	1·76	2·29	2·80	3·28	...	...	...	...	...	...	...
3¼	1·91	2·50	3·06	3·59	4·10	...	...	...	...	...	...
3½	2·07	2·71	3·32	3·91	4·47	...	...	...	...	...	...
3¾	2·23	2·92	3·58	4·22	4·83	...	...	...	...	...	...
4	2·38	3·13	3·84	4·53	5·20	5·83	...	...	...	...	...
4¼	2·54	3·33	4·10	4·84	5·56	6·25	...	...	...	...	...
4½	2·70	3·54	4·36	5·16	5·92	6·67	...	...	...	...	...
4¾	2·85	3·75	4·62	5·47	6·29	7·08	7·85	...	...	...	...
5	...	3·96	4·88	5·78	6·65	7·50	8·32	...	...	...	...
5¼	...	4·17	5·14	6·09	7·02	7·92	8·79	...	...	...	...
5½	...	4·38	5·40	6·41	7·38	8·33	9·26	10·16	...	...	...
5¾	...	...	5·66	6·72	7·75	8·75	9·73	10·68	...	...	...
6	...	...	5·92	7·03	8·11	9·17	10·20	11·20	...	...	...
6¼	...	...	6·18	7·34	8·48	9·58	10·66	11·72	12·75	...	...
6½	...	...	...	7·66	8·84	10·00	11·13	12·24	13·32	...	..
6¾	...	...	...	7·97	9·21	10·42	11·60	12·76	13·89	...	...
7	...	...	...	8·28	9·57	10·83	12·07	13·28	14·46	15·63	...
7¼	...	...	...	...	9·93	11·25	12·54	13·80	15·04	16·25	...
7½	...	...	...	...	10·30	11·67	13·01	14·32	15·61	16·88	...
7¾	...	...	...	...	10·66	12·08	13·48	14·84	16·18	17·50	18·79
8	...	...	...	...	...	12·50	13·95	15·36	16·76	18·13	19·47
8¼	...	...	...	...	...	12·92	14·44	15·89	17·33	18·75	20·14
8½	...	...	...	...	...	13·33	14·88	16·41	17·90	19·38	20·82
8¾	...	...	...	...	...	...	15·35	16·93	18·48	20·00	21·50
9	...	...	...	...	...	...	15·82	17·45	19·05	20·63	22·17
9¼	...	...	...	...	...	...	16·29	17·97	19·62	21·25	22·85
9½	...	...	...	...	...	...	...	18·49	20·20	21·88	23·53
9¾	...	...	...	...	...	...	...	19·01	20·77	22·50	24·21
10	...	...	...	...	...	...	...	19·53	21·34	23·13	24·88

Table CLI.—Weight of Steel Angle-Bars in lbs. per lineal foot.

Sum of flanges (ins.).	Thickness in fractions of an inch.										
	3/16	1/4	5/16	3/8	7/16	1/2	9/16	5/8	11/16	3/4	13/16
2½	1·47	1·91	2·32	2·71	...	...	...	...	...	...	...
2¾	1·63	2·12	2·59	3·03	...	...	...	...	..	...	...
3	1·79	2·34	2·86	3·35	...	...	...	...	...	...	...
3¼	1·95	2·55	3·12	3·67	4·18	...	...	...	...	...	...
3½	2·11	2·76	3·39	3·98	4·56	...	...	...	...	...	...
3¾	2·27	2·97	3·65	4·30	4·93	...	...	...	...	...	...
4	2·43	3·19	3·92	4·62	5·30	5·95	...	...	...	...	...
4¼	2·59	3·40	4·18	4·94	5·67	6·38	...	...	...	...	...
4½	2·75	3·61	4·45	5·26	6·04	6·80	...	...	...	...	...
4¾	2·91	3·82	4·71	5·58	6·41	7·23	8·01	...	...	...	...
5	...	4·04	4·98	5·90	6·79	7·65	8·49	...	...	...	...
5¼	...	4·25	5·25	6·22	7·16	8·08	8·96	...	...	...	...
5½	...	4·46	5·51	6·53	7·53	8·50	9·44	10·36	...	...	...
5¾	...	...	5·78	6·85	7·90	8·93	9·92	10·89	...	...	...
6	...	...	6·04	7·17	8·27	9·35	10·40	11·42	...	..	...
6¼	...	...	6·31	7·49	8·65	9·78	10·88	11·95	13·00	...	...
6½	...	...	...	7·81	9·02	10·20	11·36	12·48	13·59	...	...
6¾	...	...	...	8·13	9·39	10·63	11·83	13·02	14·17	...	...
7	...	...	...	8·45	9·76	11·05	12·31	13·55	14·76	15·94	...
7¼	...	...	...	...	10·13	11·48	12·79	14·08	15·34	16·58	...
7½	...	...	...	...	10·51	11·90	13·27	14·61	15·92	17·21	...
7¾	...	...	...	...	10·88	12·33	13·75	15·14	16·51	17·85	19·16
8	...	...	...	...	...	12·75	14·22	15·67	17·09	18·49	19·85
8¼	...	...	...	...	...	13·18	14·70	16·20	17·68	19·13	20·55
8½	...	...	...	...	...	13·60	15·18	16·73	18·26	19·76	21·24
8¾	...	...	...	...	...	...	15·66	17·27	18·85	20·40	21·93
9	...	...	...	...	...	...	16·14	17·80	19·43	21·04	22·62
9¼	...	...	...	...	...	...	16·61	18·33	20·01	21·68	23·31
9½	...	...	...	...	...	...	...	18·86	20·60	22·31	24·00
9¾	...	...	...	...	...	...	...	19·39	21·18	22·95	24·69
10	...	...	...	...	...	...	...	19·92	21·77	23·59	25·38

Table CLII.—Weights of

External Diam.	L.S.G.		THICKNESS 16	15	14	13	12	11	10	9	8	7	6	5
	Inches.		·064	·072	·080	·092	·104	·116	·128	·144	·160	·176	·192	·215
	Millimetres		1·626	1·829	2·032	2·337	2·642	2·946	3·251	3·658	4·064	4·470	4·877	5·385
Ins.	Usual L.S.G. and Weight.		WEIGHT PER											
1	14	0·771	0·627	0·700	0·771	0·875	0·976	1·074	1·169	1·291	1·407	1·519	1·624	1·749
1⅛	14	0·875	0·711	0·794	0·875	0·995	1·112	1·226	1·336	1·479	1·617	1·749	1·876	2·027
1¼	13	1·116		0·888	0·980	1·116	1·248	1·377	1·504	1·668	1·826	1·979	2·127	2·304
1⅜	13	1·236		0·982	1·085	1·236	1·384	1·529	1·671	1·856	2·036	2·210	2·378	2·582
1½	13	1·356		1·077	1·190	1·356	1·520	1·681	1·839	2·045	2·245	2·440	2·630	2·859
1⅝	13	1·477		1·171	1·294	1·477	1·656	1·833	2·007	2·233	2·455	2·671	2·881	3·137
1¾	13	1·597		1·265	1·399	1·597	1·793	1·985	2·174	2·422	2·664	2·901	3·132	3·414
1⅞	13	1·718		1·359	1·504	1·718	1·929	2·137	2·342	2·610	2·873	3·131	3·384	3·602
2	12	2·065			1·608	1·838	2·065	2·288	2·509	2·799	3·083	3·362	3·635	3·969
2⅛	12	2·201			1·713	1·959	2·201	2·440	2·677	2·987	3·292	3·592	3·886	4·247
2¼	12	2·337			1·818	2·079	2·337	2·592	2·844	3·176	3·502	3·822	4·138	4·524
2⅜	11	2·744				2·199	2·473	2·744	3·012	3·364	3·711	4·053	4·389	4·802
2½	11	2·896				2·320	2·609	2·896	3·179	3·553	3·921	4·283	4·640	5·079
2⅝	11	3·048				2·440	2·745	3·048	3·347	3·741	4·130	4·514	4·892	5·357
2¾	11	3·200				2·561	2·882	3·200	3·514	3·930	4·339	4·744	5·143	5·634
2⅞	11	3·351				2·681	3·018	3·351	3·682	4·118	4·549	4·974	5·394	5·912
3	11	3·503				2·802	3·154	3·503	3·850	4·307	4·758	5·205	5·646	6·189
3⅛	10	4·017					3·290	3·655	4·017	4·495	4·968	5·435	5·897	6·467
3¼	10	4·185					3·426	3·807	4·185	4·684	5·177	5·665	6·148	6·744
3⅜	10	4·352					3·562	3·959	4·352	4·872	5·387	5·896	6·400	7·022
3½	10	4·520	This table of weights is based on the specific gravity of Iron at 7·698. Actual weights may differ slightly.				3·698	4·111	4·520	5·061	5·596	6·126	6·651	7·299
3⅝	10	4·687					3·835	4·262	4·687	5·249	5·806	6·357	6·902	7·577
3¾	10	4·855					3·971	4·414	4·855	5·438	6·015	6·587	7·154	7·854
3⅞	9	5·626						4·566	5·022	5·626	6·224	6·817	7·405	8·132
4	9	5·815						4·718	5·190	5·815	6·434	7·048	7·656	8·410

For other metals—

Steel,	.. ..	multiply by 1·012.
Brass,	.. ..	,, 1·080
Copper,	.. ..	,, 1·135

Iron Boiler Tubes.

OF IRON.											
4	3	2	1	½"	5/16"	¼"	3/16"	⅛"	1/10"	1/16"	External Diam.
·232	·252	·276	·300	·125	·187	·250	·313	·375	·437	·500	
5·893	6·401	7·010	7·620	3·175	4·726	6·350	7·937	9·525	11·112	12·700	
FOOT IN POUNDS.											Ins.
1·366	1·974	2·092	2·199	1·145	1·595	1·963	2·250	2·454	2·577	2·618	1
2·169	2·304	2·454	2·592	1·309	1·841	2·291	2·659	2·945	3·150	3·272	1¼
2·473	2·634	2·815	2·984	1·473	2·086	2·618	3·068	3·436	3·723	3·927	1½
2·777	2·963	3·176	3·377	1·636	2·332	2·945	3·477	3·927	4·295	4·581	1¾
3·081	3·293	3·538	3·770	1·800	2·577	3·272	3·886	4·418	4·868	5·236	1¾
3·384	3·623	3·899	4·163	1·963	2·822	3·600	4·295	4·909	5·440	5·890	1⅞
3·688	3·953	4·260	4·555	2·127	3·068	3·927	4·704	5·400	6·013	6·545	1¾
3·992	4·283	4·621	4·948	2·291	3·313	4·254	5·113	5·890	6·586	7·199	1¾
4·295	4·613	4·983	5·341	2·454	3·559	4·581	5·522	6·381	7·159	7·854	2
4·599	4·943	5·344	5·733	2·618	3·804	4·909	5·931	6·872	7·731	8·508	2⅛
4·903	5·273	5·705	6·126	2·782	4·050	5·236	6·340	7·363	8·304	9·163	2¼
5·206	5·602	6·067	6·519	2·945	4·295	5·563	6·749	7·854	8·877	9·817	2⅜
5·510	5·932	6·428	6·911	3·109	4·541	5·890	7·159	8·345	9·449	10·472	2½
5·814	6·262	6·789	7·304	3·272	4·786	6·218	7·568	8·836	10·022	11·126	2⅝
6·117	6·592	7·150	7·697	3·436	5·031	6·545	7·977	9·327	10·595	11·781	2¾
6·421	6·922	7·512	8·090	3·600	5·277	6·872	8·386	9·818	11·167	12·435	2⅞
6·725	7·252	7·873	8·482	3·763	5·522	7·200	8·795	10·308	11·740	13·090	3
7·028	7·582	8·234	8·875	3·927	5·768	7·527	9·204	10·799	12·313	13·744	3⅛
7·332	7·911	8·596	9·268	4·091	6·013	7·854	9·613	11·290	12·885	14·399	3¼
7·636	8·241	8·957	9·660	4·254	6·259	8·181	10·022	11·781	13·458	15·053	3⅜
7·940	8·571	9·318	10·053	4·418	6·504	8·508	10·431	12·272	14·031	15·708	3½
8·243	8·901	9·679	10·446	4·581	6·750	8·836	10·840	12·763	14·604	16·362	3⅝
8·547	9·231	10·041	10·838	4·745	6·995	9·163	11·249	13·254	15·176	17·017	3¾
8·851	9·561	10·402	11·231	4·908	7·240	9·490	11·658	13·745	15·749	17·671	3⅞
9·154	9·891	10·763	11·624	5·072	7·486	9·817	12·067	14 ·235	16·322	18·326	4

Table CLIII.—Standard List of Iron Welded Tubes as supplied by Tube Makers.

External Diameter.	List Particulars. Thickness. L.S.G.	Thickness. Inch.	Price per ft.	Per Foot. Surface. sq. ft.	Weight. lbs.
			d.		
1	14	·080	..	·262	0·771
1 1/8	14	·080	..	·295	0·875
1 1/4	13	·092	9·5	·327	1·116
1 3/8	13	·092	9·5	·360	1·236
1 1/2	13	·092	9·5	·393	1·356
1 5/8	13	·092	9·75	·425	1·477
1 3/4	13	·092	10·0	·458	1·597
1 7/8	13	·092	10·25	·491	1·718
2	12	·104	11·75	·523	2·065
2 1/8	12	·104	12·25	·556	2·201
2 1/4	12	·104	12·25	·589	2·337
2 3/8	11	·116	14·0	·622	2·744
2 1/2	11	·116	14·5	·654	2·896
2 5/8	11	·116	15·25	·687	3·048
2 3/4	11	·116	15·75	·720	3·200
2 7/8	11	·116	16·50	·752	3·351
3	11	·116	17·0	·785	3·503
3 1/4	10	·128	20·0	·851	4·185
3 1/2	10	·128	22·0	·916	4·520
3 3/4	10	·128	23·5	·982	4·855
4	9	·144	28·5	1·047	5·815
4 1/4	9	·144	31·5	1·112	6·192
4 1/2	9	·144	33·5	1·178	6·569
4 3/4	8	·160	40·5	1·243	7·691
5	8	·160	43·5	1·309	8·109
5 1/4	8	·160	47·0	1·376	8·528
5 1/2	7	·176	54·0	1·442	9·813
5 3/4	7	·176	59·0	1·507	10·27
6	7	·176	62·0	1·572	10·73
6 1/4	7	·176	70·0	1·638	11·20
6 1/2	7	·176	73·0	1·703	11·66
6 3/4	7	·176	79·0	1·769	12·12
7	7	·176	82·0	1·834	12·58
7 1/4	5	·212	105	1·900	15·63
7 1/2	5	·212	109	1·965	16·18
7 3/4	5	·212	113	2·030	16·74
8	5	·212	117	2·093	17·29
8 1/4	3	·252	147	2·159	21·11
8 1/2	3	·252	151	2·225	21·77
8 3/4	3	·252	156	2·291	22·43
9	3	·252	160	2·357	23·09
9 1/4	3	·252	175	2·423	23·75
9 1/2	3	·252	182	2·490	24·41
9 3/4	3	·252	186	2·556	25·06
10	3	·252	192	2·618	25·72
10 1/2	2	·276	216	2·683	28·83
10 3/4	2	·276	222	2·749	29·55
10 3/4	1	·300	246	2·814	32·33
11	1	·300	252	2·880	33·59
11 1/4	1	·300	258	2·946	34·36
11 1/2	1	·300	264	3·012	35·12
11 3/4	1	·300	270	3·078	35·88
12	1	·300	276	3·142	36·65
12 1/4	1	·300	288	3·273	38·18
12 1/2	1	·300	300	3·403	39·70
13	1	·300	312	3·534	41·23
13 1/2	1	·300	324	3·665	42·76
14					

Note.—The above prices are for iron or steel lap-welded tubes, and are subject to a trade discount varying from 60 per cent. in good times to 75 in bad. The weight of steel tubes will be 2·88 per cent. or $^{1}/_{42}$ heavier than those given above. *Welded steel tubes* can be got at about the same rates as iron ones.

Table CLIV.—Weight of Large Steel Tubes Solid Drawn or Welded, in lbs. per foot.

External diameter.	Thickness in 64ths of an inch.							External diameter.	Thickness in 64ths of an inch.						
	8	9	10	11	12	14	16		14	16	18	20	22	24	26
ins.								ins.							
10	13·19	14·81	16·43	18·05	19·65	22·86	26·04	25	57·91	66·09	74·26	82·41	90·53	98·64	106·71
11	14·52	16·31	18·10	19·88	21·66	25·19	28·71	26	60·24	68·76	77·27	85·75	94·20	102·64	111·06
12	15·86	17·81	19·77	21·72	23·66	27·53	31·78	27	62·58	71·43	80·27	89·08	97·87	106·65	115·40
13	17·19	19·32	21·44	23·55	25·66	29·87	34·05	28	64·92	74·11	83·27	92·42	101·55	110·65	119·74
14	18·53	20·82	23·11	25·39	27·66	32·20	36·72	29	...	76·78	86·28	95·76	105·22	114·66	124·08
15	19·86	22·32	24·77	27·22	29·67	34·54	39·39	30	...	79·45	89·28	99·10	108·89	118·66	128·42
16	21·20	23·82	26·44	29·06	31·67	36·88	42·06	31	...	82·12	92·29	102·44	112·56	122·67	132·76
17	...	25·33	28·11	30·90	33·67	39·21	44·73	32	...	84·79	95·29	105·77	116·23	126·67	137·10
18	...	26·83	29·78	32·73	35·68	41·55	47·40	33	...	...	98·29	109·11	119·91	130·68	141·43
19	...	...	31·45	34·57	37·68	43·88	50·07	34	...	...	101·30	112·45	123·58	134·68	145·78
20	...	...	33·12	36·41	39·68	46·22	52·74	35	...	...	104·30	115·79	127·25	138·69	150·11
21	...	...	...	38·24	41·68	48·56	55·41	36	...	...	107·31	119·13	130·92	142·70	154·45
22	...	...	...	40·08	43·69	50·89	58·08	37	...	...	110·31	122·47	134·59	146·70	158·79
23	...	...	...	...	45·69	53·23	60·75	38	...	...	113·32	125·81	138·26	150·71	163·13
24	...	...	...	...	49·70	55·57	63·42	39	...	...	116·33	129·15	141·93	154·72	167·47

N.B.—Iron tubes will be $^{1}/_{42}$ lighter, that is $\frac{41}{42}$, or 0·976 of the above.

Table CLV.—Weights of Seamless

L. S. G. .	4/0	3/0	2/0	0	1	2	3	4	5
Inches .	·400	·372	·348	·324	·300	·276	·252	·232	·212
Millimetres .	10·160	9·449	8·839	8·229	7·620	7·010	6·401	5·893	5·385
Internal Diameter.	Weight of a lineal								
Inches / mm.									
¾ 19·0	...	...	...	...	...	...	...	...	...
1 25·4	...	...	...	...	...	...	...	...	...
1¼ 31·7	...	...	...	...	...	...	...	...	...
1½ 38·1	...	...	...	...	...	...	...	...	...
1¾ 44·4	...	...	...	...	...	...	...	...	...
2 50·8	...	...	...	...	...	...	...	...	...
2¼ 57·1	...	...	...	...	...	...	...	...	...
2½ 63·5	...	...	...	...	...	...	...	...	...
2¾ 69·8	...	...	...	...	...	...	...	...	7·60
3 76·2	...	...	...	...	...	...	...	9·07	8·24
3¼ 82·5	...	...	...	...	...	...	...	9·77	8·88
3½ 88·9	...	...	...	...	...	...	11·44	10·47	9·52
3¾ 95·2	...	...	...	...	...	13·44	12·20	11·18	10·16
4 101·6	...	...	...	...	...	14·28	12·96	11·88	10·80
4½ 114·3	...	...	...	...	17·42	15·95	14·49	13·28	12·08
5 127·0	...	...	...	20·87	19·23	17·62	16·01	14·68	13·37
5½ 139·7	...	...	...	22·83	21·05	19·29	17·54	16·09	14·65
6 152·4	...	...	26·72	24·79	22·86	20·95	19·06	17·49	15·93
6½ 165·1	...	30·93	28·83	26·75	24·68	22·62	20·58	18·89	17·21
7 177·8	...	33·18	30·93	28·71	26·49	24·29	22·11	20·30	18·50
7½ 190·5	38·23	35·43	33·04	30·67	28·31	25·96	23·63	21·70	19·78
8 203·2	40·65	37·68	35·14	32·63	30·12	27·63	25·16	23·10	21·06
8½ 215·9	43·07	39·93	37·25	34·59	31·94	29·30	26·68	24·51	22·34
9 228·6	45·49	42·18	39·35	36·55	33·75	30·97	28·21	25·91	23·63
9½ 241·3	47·91	44·43	41·46	38·51	35·57	32·64	29·73	27·31	24·91
10 254·0	50·33	46·68	43·56	40·47	37·38	34·31	31·25	28·72	26·19
10½ 266·7	52·75	48·93	45·67	42·43	39·20	35·98	32·78	30·12	27·47
11 279·4	55·16	51·18	47·77	44·39	41·01	37·65	34·30	31·52	28·75
11½ 292·1	57·58	53·43	49·88	46·35	42·83	39·32	35·83	32·93	30·04
12 304·8	60·00	55·68	51·98	48·31	44·64	40·99	37·35	34·33	31·32
12½ 317·5	62·42	57·93	54·09	50·26	46·45	42·66	38·88	35·73	32·60
13 330·2	64·84	60·18	56·19	52·22	48·27	44·33	40·40	37·14	33·88
13½ 342·9	67·26	62·43	58·30	54·18	50·08	46·00	41·92	38·54	35·17
14 355·6	69·68	64·68	60·40	56·14	51·90	47·67	43·45	39·94	36·45

Mandril drawn Brazed Tubes weigh the

Ordinary Brazed Copper Tubes weigh rather more than Seamless Tubes, the

Copper Tubes.

6	7	8	9	10	11	12	13	14	15	16
·192	·176	·160	·144	·128	·116	·104	·092	·080	·072	·064
4·877	4·470	4·064	3·658	3·251	2·946	2·642	2·377	2·032	1·829	1·626
foot in pounds.										
...	...	...	...	1·86	1·21	1·07	·94	·80	·72	·63
...	...	...	...	1·75	1·57	1·39	1·21	1·04	·93	·82
...	...	...	2·48	2·13	1·92	1·70	1·49	1·29	1·15	1·02
...	...	3·21	2·86	2·52	2·27	2·02	1·77	1·53	1·37	1·21
...	...	3·70	3·30	2·91	2·62	2·33	2·05	1·77	1·59	1·40
...	4·63	4·18	3·73	3·29	2·97	2·65	2·33	2·01	1·80	1·60
5·67	5·16	4·66	4·17	3·68	3·32	2·96	2·61	2·25	2·02	1·79
6·25	5·70	5·15	4·61	4·07	3·67	3·28	2·88	2·50	2·24	1·98
6·83	6·23	5·63	5·04	4·46	4·02	3·59	3·16	2·74	2·46	2·18
7·41	6·76	6·12	5·48	4·84	4·37	3·90	3·44	2·98	2·68	2·37
7·99	7·29	6·60	5·91	5·23	4·72	4·22	3·72	3·22	2·89	2·57
8·58	7·83	7·08	6·35	5·62	5·07	4·53	4·00	3·46	3·11	2·76
9·16	8·36	7·57	6·78	6·00	5·42	4·85	4·28	3·71	3·33	2·95
9·74	8·89	8·05	7·22	6·39	5·78	5·16	4·55	3·95	3·55	3·15
10·90	9·96	9·02	8·09	7·17	6·48	5·79	5·11	4·43	3·98	3·53
12·06	11·02	9·99	8·96	7·94	7·18	6·42	5·67	4·92	4·42	3·92
13·22	12·08	10·96	9·83	8·71	7·88	7·05	6·22	5·40	4·85	4·31
14·38	13·15	11·92	10·70	9·49	8·58	7·68	6·78	5·88	5·29	4·69
15·54	14·21	12·89	11·57	10·26	9·28	8·31	7·34	6·37	5·72	5·08
16·70	15·28	13·86	12·44	11·04	9·99	8·94	7·89	6·85	6·16	5·47
17·87	16·34	14·83	13·32	11·81	10·69	9·57	8·45	7·34	6·60	5·86
19·03	17·41	15·79	14·19	12·59	11·39	10·20	9·01	7·82	7·03	6·24
20·19	18·47	16·76	15·06	13·36	12·09	10·82	9·56	8·30	7·47	6·63
21·35	19·54	17·73	15·93	14·13	12·79	11·45	10·12	8·79	7·90	7·02
22·51	20·60	18·70	16·80	14·91	13·49	12·08	10·68	9·27	8·34	...
23·67	21·67	19·67	17·67	15·68	14·20	12·71	11·23	9·76	8·77	...
24·83	22·73	20·63	18·54	16·46	14·90	13·34	11·79	10·24	...	...
26·00	23·80	21·60	19·41	17·23	15·60	13·97	12·34	10·72	...	...
27·16	24·86	22·57	20·28	18·01	16·30	14·60	12·90	...	...	...
28·32	25·92	23·54	21·16	18·78	17·00	15·23	13·46	...	...	...
29·48	26·99	24·50	22·03	19·55	17·70	15·86	...	...	...	...
30·64	28·05	25·47	22·90	20·33	18·41	16·49	...	...	...	...
31·80	29·12	26·44	23·77	21·10	19·11	...	...	...	...	...
32·96	30·18	27·41	24·64	21·88	19·81	...	...	...	...	...

same as Seamless Tubes.
difference varying with the thickness, diameter, and class of joint.

Table CLVI.—Whitworth Gas Threads.

Diameter of pipe (inside).	Diameter over threads.	Diameter at bottom of thread.	Number of threads per inch.
1/8"	·3825	·3367	28
1/4	·5180	·4506	19
3/8	·6563	·5889	19
1/2	·8257	·7342	14
5/8	·9022	·8107	14
3/4	1·041	·9495	14
7/8	1·189	1·0975	14
1	1·309	1·1925	11
1 1/8	1·492	1·3755	11
1 1/4	1·65	1·5335	11
1 3/8	1·745	1·6285	11
1 1/2	1·8825	1·766	11
1 5/8	2·021	1·9045	11
1 3/4	2·047	1·9305	11
1 7/8	2·245	2·1285	11
2	2·347	2·2305	11
2 1/4	2·5875	2·471	11
2 1/2	3·0013	2·8848	11
2 3/4	3·247	3·1305	11
3	3·485	3·3685	11
3 1/4	3·6985	3·582	11
3 1/2	3·912	3·7955	11
3 3/4	4·1225	4·009	11
4	4·339	4·2225	11

Table CLVII.—Weight of Brass Condenser Tubes.

L.S.G.		14	15	16	17	18	19	20
Inches.		·080	·072	·064	·056	·048	·040	·036
Millimetres.		2·032	1·829	1·626	1·422	1·219	1·016	·914
Ext. Diameter.		Weight of a Lineal Foot in pounds.						
Ins.	mm.							
1/2	15·9	·51	·46	·42	·37	·32	·27	·25
3/4	19·0	·62	·57	·51	·45	·39	·33	·30
7/8	22·2	74	·67	·60	·53	·46	·39	·35
1	25·4	·86	·78	·70	·62	·53	·45	·40

Note.—The above weights are for tubes containing 70 per cent. copper.

Table CLVIII.—Weight of Lead Pipes.

Thickness in inches.	Weight in pounds per foot run.										
	2″	2¼″	2½″	2¾″	3″	3¼″	3½″	3¾″	4″	4½″	5″
¼	8·7	9·7	10·6	11·6	12·6	13·5	14·5	15·5	16·4	18·4	20·3
5/16	...	...	13·6	14·8	16·0	17·2	18·4	19·7	20·9	23·3	25·7
⅜	...	...	...	...	...	21·0	22·5	23·9	25·4	28·3	31·2
7/16	...	...	...	...	...	...	...	...	...	33·4	36·8

Lead pipes of these sizes and weights are usually manufactured in 10 ft. lengths.

Table CLIX.—Weight of Sheet Metals.

Thickness L.S.G.	Weight in pounds per square foot.					
	Steel.	Iron.	Copper.	Brass.	Lead.	Zinc.
7/0	20·40	20·00	22·83	21·98	29·65	18·72
6/0	18·93	18·56	21·19	20·39	27·51	17·38
5/0	17·63	17·28	19·73	18·99	25·62	16·18
4/0	16·32	16·00	18·27	17·58	23·72	14·98
3/0	15·18	14·88	16·99	16·35	22·06	13·93
2/0	14·20	13·92	15·89	15·30	20·64	13·03
0	13·22	12·96	14·80	14·24	19·21	12·13
1	12·24	12·00	13·70	13·19	17·79	11·23
2	11·26	11·04	12·60	12·13	16·37	10·34
3	10·28	10·08	11·51	11·08	14·94	9·44
4	9·47	9·28	10·59	10·20	13·76	8·69
5	8·65	8·48	9·68	9·32	12·57	7·94
6	7·83	7·68	8·77	8·44	11·38	7·19
7	7·18	7·04	8·04	7·74	10·44	6·59
8	6·53	6·40	7·31	7·03	9·49	5·99
9	5·88	5·76	6·58	6·33	8·54	5·39
10	5·22	5·12	5·84	5·63	7·59	4·79
11	4·73	4·64	5·30	5·10	6·88	4·34
12	4·24	4·16	4·75	4·57	6·17	3·89
13	3·75	3·68	4·20	4·04	5·45	3·44
14	3·26	3·20	3·65	3·52	4·74	3·00
15	2·94	2·88	3·29	3·16	4·27	2·70
16	2·61	2·56	2·92	2·81	3·79	2·40
17	2·28	2·24	2·56	2·46	3·32	2·10
18	1·96	1·92	2·19	2·11	2·85	1·80
19	1·63	1·60	1·83	1·76	2·37	1·50
20	1·47	1·44	1·64	1·58	2·13	1·35
21	1·31	1·28	1·46	1·41	1·90	1·20
22	1·14	1·12	1·28	1·23	1·66	1·05

Table CLX.—Weights of Engines and Boilers, &c.

Description of Ship and Machinery.	Weight in Pounds per H.P.			I.H.P. Per Ton Total Weight.
	Engines, Propellers, &c.	Boilers, &c.	Total Machinery.	
(1) Battleships and large cruisers, triple engines, tank boilers, 155 lbs. pressure	115	150	265	8·45
(2) Battleships and large cruisers, triple engines, water tube boilers, 250 lbs. pressure	115	131	246	9·11
(3) Battleships and large cruisers, turbines, mixed boilers, 215 lbs. pressure	..	..	177	12·68
(4) Large high-speed cruisers, triples, tank boilers, 155 lbs., M.F.D.	..	..	202	11·09
(5) Large high-speed cruisers, triple, water tube boilers, 250 lbs., M.F.D.	..	..	188	11·92
(6) Large high-speed cruisers, triples, mixed boilers, 210 lbs., M.F.D.	..	..	215	10·42
(7) Large high-speed cruisers, turbines, water-tube boilers, 235 lbs. M.F.D., oil-fired	..	..	84	30·3
(8) Moderate size cruisers, with triple engines, tank boilers, 155 lbs., M.F.D.	..	..	195	11·5
(9) Moderate size cruisers, with triple engines water-tube boilers, 250 lbs., M.F.D.	..	..	185	12·1
(10) Moderate size cruisers, with turbines, water tube boilers, 210 lbs., M.F.D.	..	..	91·8	24·2
(11) T.B.D. leaders, geared turbines, water-tube boilers, 250 lbs. pressure	..	..	32	7·0
(12) Scouts, &c., turbines, water-tube boilers	..	..	59·7	37·5
(13) Destroyers, triple engines, water-tube boilers	24·0	24·0	48·0	45·0
(14) Destroyers, turbines, water-tube boilers	16·4	21·6	37·8	59·3
(15) Largest and fastest express Atlantic steamers, reciprocators, tank boilers	..	..	345	6·49
(16) Largest and fastest express Atlantic steamers, turbines, tank boilers	..	..	307	7·28
(17) Large expresses: 20 knots, passenger and cargo, Atlantic, reciprocators	..	..	440	5·09
(18) Large expresses: 20 knots, passenger and cargo, Atlantic, turbines	202	223	425	5·25
(19) Express channel and excursion steamers, reciprocators	..	..	213	10·52
(20) Express channel and excursion steamers, turbines	47	121	168	13·33
(21) Large cargo and passenger steamers, Eastern service, reciprocators	..	..	330	6·79
(22) Ordinary cargo steamers, triples and quadruples	..	..	462	4·85
(23) Large express short-service paddle steamers, compound diagonal engines	..	..	285	7·86
(24) Ordinary river service, moderate size, compound diagonal engines	185	180	365	6·14

The machinery of torpedo boats with loco. boilers weighed about 30 to 35 lbs. per I.H.P. in the engine-room, and about 45 to 50 lbs. in the boiler-room, or say 81 lbs. per I.H.P. for both engine- and boiler-room. When water-tube boilers are used, the boiler-room weights are reduced to about 33 lbs. per I.H.P. Now with impulse turbines, with single-geared reduction and Yarrow type of water-tube boilers with superheaters the total weight of machinery of T.B.Ds. with steam up is only 32 lbs. per I.H.P. That of some of the later 30 knot T.B.Ds., was as low as 43 lbs. with reciprocating engines.

Modern large cruisers with impulse turbines and single reduction and small tube Yarrow boilers have machinery whose total weight per I.H.P. is only 50 lbs. which is about the same as the light cruisers with direct driven turbines.

The machinery of the Naval pinnaces was from 90 to 115 lbs. per I.H.P., and that of the cutters 150 to 190 lbs. With the adoption of water-tube boilers in the place of locomotive boilers in these crafts, these weights have been considerably reduced.

The weights of paddle engines depend to some extent on the service for which the vessel is intended. For open sea and cross-channel work the wheels, etc., must be considerably heavier than for those in smooth water service or simple coasting.

CENTRE OF GRAVITY OF MACHINERY, &c.

When the common centre of gravity of a number of detached bodies, —such as the various portions of the machinery of a steamship,—is required, it is obtained as follows:—

1. **Longitudinal Position of Centre of Gravity.**—On the longitudinal section drawing of the machinery, near where it is supposed the longitudinal position of the centre of gravity will be, draw a vertical line, to represent the edge of a plane cutting the ship transversely. Then, take the weight of each portion of the machinery, and multiply it by the distance of its centre of gravity from this transverse plane; distinguish the resulting quantities, or moments, by the signs + and −, according as they happen to be derived from the right or left side of the plane; and add together all the plus quantities, and all the minus quantities. Then obtain the resultant, by subtracting the less quantity from the greater, and place before it the sign of the greater.

If this resultant moment be now divided by the *total* weight of the whole system of bodies, on both sides of the plane, the quotient will be the distance of the longitudinal position of the centre of gravity from the plane; and it will lie to the right, or to the left, of the plane, according as the sign of the quotient is plus or minus.

2. **Transverse Position of Centre of Gravity.**—Assume another plane, perpendicular to the first, and represent it by a line on the transverse section drawings, and proceed exactly as described above.

3. **Vertical Position of Centre of Gravity.**—Assume a third plane, perpendicular to both those previously used (parallel to the water-line, that is), represent it by a line or lines as before, and again calculate the position as described above.

MISCELLANEOUS TABLES.

Table CLXI.—Surface of Tubes in square feet per foot run.

Diamr. in inches.	0	⅛	¼	⅜	½	⅝	¾	⅞
0	...	·0327	·0654	·0982	·1309	·1636	·1963	·2291
1	·2618	·2945	·3272	·3600	·3927	·4254	·4581	·4909
2	·5236	·5563	·5890	·6218	·6545	·6872	·7200	·7527
3	·7854	·8181	·8508	·8836	·9163	·9490	·9817	1·0145
4	1·0472	1·0799	1·1126	1·1454	1·1781	1·2108	1·2435	1·2763
5	1·3090	1·3417	1·3744	1·4072	1·4399	1·4726	1·5053	1·5381

Bulk and Weight, &c., of Water.

Table CLXII.—Fresh Water.

Cubic inches.	Cubic feet.	Pounds.	Gallons.	Ton.
1	...	·036	...	...
27·7	·016	1·0	·1	...
277	·160	10·0	1·0	·00446
1728	1·0	62·4	6·24	·0279
...	35·9	2240	224	1·0

The above figures are very nearly correct for 62° F.: for bulk of water at other temperatures see page 400.

Table CLXIII.—Salt Water.

Cubic inches.	Cubic feet.	Pounds.	Gallons.	Ton.
1	...	·037	...	...
27	·0156	1·0	·097	...
277	·160	10·26	1·0	·00458
1728	1·0	64·0	6·24	·0285
...	35·0	2240	218·3	1·0

The above figures are very nearly correct for 62° F. : for bulk of water at other temperatures see page 400.

Table CLXIV.—Solid Matter deposited from Feed Water—

Prof. V. Lewes.

Constituents.	River.	Tidal Rivers.	Sea.
Calcic carbonate, . . .	75·85	43·65	0·97
,, sulphate, . . .	3·68	34·78	85·53
Magnesic hydrate, . . .	2·56	4·34	3·39
Sodic chloride, ...	0·45	0·56	2·79
Silica, ...	7·66	7·52	1·10
Oxides of iron and alumina, .	2·96	3·44	0·32
Organic matter, . . .	3·64	1·55	trace
Moisture, ...	3·20	4·16	5·90
	100·00	100·00	100·00

Table CLXV.—Sea-Water, Solid Contents of.

Sea water contains certain mineral salts in solution amounting to 3·4 per cent. of its weight on the average, and is at the various parts of the globe as follows :—

Arctic Ocean .	2·83 per cent.	Red Sea . .	4·30 per cent
North Atlantic .	4·26 ,,	Dead Sea . .	38·66 ,,
South ,, .	4·12 ,,	English Channel	3·55 ,,
Equator . .	3·94 ,,	Irish ,,	3·38 ,,
Mediterranean .	3·94 ,,	Upper Baltic .	0·66 ,,
Sea of Marmora .	4·20 ,,	Cronstadt . .	2·00 ,,
Black Sea . .	2·16 ,,		

Table CLXVI.—Composition of Solids in Solution in Sea-Water.

Salts in Sea Water.	Kingston Harbour. Main and Brown.	Average. D. K. Clerk.	English Channel. Thomson.
Chloride of sodium, . .	71·32	73·60	76·70
Chloride of potassium, .	—	—	2·17
Chloride of magnesium, .	10·79	9·80	10·40
Sulphate of magnesium, .	5·30	15·73	6·56
Sulphate of lime, . .	4·87	0·29	4·0
Carbonate of lime, . .	1·73	0·58	0·09
Bromide of magnesium, .	0·60	—	0·08
Organic matter, . .	5·27	—	—

Table CLXVII.—Weight of Sea-Water at various Ports and Harbours of the United Kingdom, taken at high-water and given in ounces per cubic foot; fresh-water being 1000.

Locality.	Weight	Locality.	Weight
Thames at Limehouse .	1000	Plymouth Sound . .	1025
Do. Victoria Dock .	1005	Weymouth . . .	1024
Do. Gravesend .	1019	Southampton Extension Dock	1023
Humber at Goole Dock .	1000	Portsmouth, Camber. .	1024
Do. off Hull . .	1016	Dublin	1024
Do. off Grimsby .	1020	Kingstown . . .	1025
Tyne at Low Walker .	1013	Queenstown Harbour .	1025
Do. Tyne Dock . .	1023	Cork	1000
Tees at Stockton . .	1005	Youghal	1025
Do. Middlesbrough .	1012	Limerick	1000
Do. Hartlepool . .	1025	Belfast Harbour . .	1011
Sunderland Harbour .	1024	Glasgow, Bowling . .	1000
Mersey at Connahs Quay .	1018	Greenock, Tail of Bank .	1022
Do. Liverpool . .	1021	Port Glasgow, Dock Gates	1013
Milford Haven . . .	1023	Troon	1023
Avonmouth Dock, Bristol.	1011	Campbeltown . . .	1024
Portishead Dock, do. .	1015	Fraserburgh Harbour .	1025
Cardiff, Roath Basin. .	1008	Peterhead Harbour . .	1024
Do. Penarth Basin .	1018	Dundee	1021
Barrow, Buccleugh Dock .	1017	St Andrews . . .	1025
Whitehaven, Queen's Dock	1024	Leith and Granton, Forth at	1025
Birkenhead Float . .	1020	Bo'ness Dock . . .	1021

Table CLXVIII.—Boiling Points, &c., of Sea-Water.

Proportion of salt in the water.	Temperature, Fahrenheit, at which it boils.	Specific Gravity.
1/32	213·2°	1·029
2/32	214·4°	1·058
3/32	215·5°	1·087
4/32	216·7°	1·116
5/32	217·9°	1·145
6/32	219·1°	1·174
7/32	220·3°	1·203
8/32	221·5°	1·232
9/32	222·7°	1·261
10/32	223·8°	1·290
11/32	225·0°	1·319
12/32	226·1°	1·348

Note.—Freezing Mixtures are made as follows :—(*a*) Mix one part of common salt with two parts of pounded ice ; the temperature will be – 4° F. or – 20° C. ; (*b*) mix three parts of chloride of calcium with two parts of snow ; the temperature then is – 50° F. or – 45·5° C.

Table CLXIX.—Viscosity of Oils at Various Temperatures (Water at 68° F. being 0·01028)—*Archbutt and Deeley.*

Name of Oil.	60° F.	100° F.	150° F.	212° F.
Sperm oil,	0·420	0·185	0·085	0·046
Olive ,,	1·008	0·377	0·154	0·070
Rape or colza oil,	1·118	0·422	0·177	0·080
Castor oil,	...	2·729	0·605	0·169
Scotch mineral oil "865,"	0·146	0·066	0·036	...
,, ,, "890,"	0·509	0·183	0·069	...
Russian light machinery oil,	1·156	0·307	0·099	0·043
,, heavy ,,	3·592	0·762	0·196	0·066
American spindle oil,	0·727	0·236	0·086	0·039
,, pale ,, "903/7,"	1·138	0·342	0·115	0·049
,, ,, "907/12,"	1·479	0·419	...	...
,, red engine oil "910/20,"	1·915	0·496	0·150	0·058
,, "Bayonne" engine oil "912/15,"	2·172	0·572	0·173	0·063
,, dark medium machinery oil,	3·046	0·705	0·210	0·076
"Valvoline,"	...	2·406	0·605	0·187
American filtered cylinder oil "888,"	...	3·702	...	0·238
,, dark ,, ,, "900,"	...	6·264	...	0·314

Table CLXX.

The oils generally used in connection with ordinary machinery will give satisfactory results if they comply with the following specifications:—

Items.	Specific Gravity at 60° F.	Viscosity at 140° F. Secs.	Viscosity at 200° F. Secs.	Viscosity at 300° F. Secs.	Flashing Point.	Loss at 212° F. in 2 hours not more than	Saponifiable Oil %	Free Acid not exceeding
*Machinery oil (ordinary)	Abt. 0·90	165–180	60–65	—	Not below 370° F.	0·15 %	10–12 %	0·2 %
Machinery oil for forced lubrication	Abt. 0·91	Abt. 85	Abt. 44·5	Abt. 31	390°–400° F.	0·2 %	Nil.	0·1 %
†Cylinder oil . . .	Abt. 0·90	180–200	65–70	35–40	Not under 395° F.	0·1 %	5–6 %	0·1 %
*Gas engine oil . . .	Abt. 0·91	82–87	44–48	32–34	,,	0·2 %	5–6 %	0·1 %
For heavy or troublesome bearings either pure castor oil or oil to the following specification may be used.	Abt. 0·91	180–200	65–70	—	Not below 350° F.	0·2 %	18–22 %	0·4 %

NOTES.

The gas engine oil will also serve very well for light high-speed machinery.

Viscosities to be taken with Boverton Redwood's apparatus.

Flash points to be determined by means of the Pensky-Martens apparatus.

The hydrocarbon oils should be free from grit or dirt, acid, alkali, resin oil, bituminous matter, and all artificial thickening agents; they should also form a clear mixture with rape oil at ordinary temperatures. The saponifiable oil should be lard or tallow oil (not any seed oil), and should not contain more than 2 per cent. of free fatty acid.

The hydrocarbon oils, after being kept at 25° F. for one hour in a ½-inch. int. dia. test tube, should flow when the tube is inclined.

* These oils should be either distilled or charcoal refined or a blend of these types with the addition of the necessary percentage of pure lard or tallow oil only (no seed oil should be accepted), and should be practically transparent.

† A proportion of "natural" oil may be used in producing the necessary blends, but the oil supplied must be free from bituminous matter.

As viscosity at 300 F. is troublesome to ascertain, the oil may, as an alternative test, be kept at 365° F. for 24 hours and should not suffer any appreciable change in properties, though it may be slightly darkened in colour.

Table CLXXI.—Boiling, Setting, and Flash Points.

Name or Brand of Oil.	Sets at	Flashes at	Boils at
	Degrees F.	Degrees F.	Degrees F.
American cylinder, sp. gr. ·902,	25	585	650
,, ,, ,, ·893,	25	424	600
Valvoline ,,	...	...	700
American machine, sp. gr. ·897,	25	402	600
Russian ,, ,, ·909,	0 to 10	380	...
Scotch shale oil,	32	...	...
Cotton-seed oil,	34 to 50	...	580
Castor oil,	14	...	556
Lard oil,	25 to 42	...	...
Linseed oil,	5 ,, 17	...	574
Neat's foot oil,	32 ,, 50	...	...
Olive oil,	21 ,, 39	...	...
Rape or colza oil,	10 ,, 28	...	625
Sperm oil,	32	450	600
Palm oil,	81 to 106	...	...
Whale oil,	32	440	620
Vaseline oil,	104 to 120	640	...
Tallow (Russian),	107 ,, 119	540	660

Table CLXXII.—Co-efficients of Friction.

Nature of the Surfaces.	Quite Dry.	Greasy Dry.	Well Lubricated with			Constant Flow of Oil.
			Oil.	Tallow or Soap.	Water.	
Wood on wood (dry),	0·25 to 0·5	...	...	0·2 to 0·04	...	...
Metal on hard wood,	...	0·15	0·07 to 0·08	...	...	0·05
Metals on oak, (Rk)	0·5 to 0·6	0·24 to 0·26	...	0·2	...	...
,, metals only, ,,	...	0·15 to 0·2	...	0·14	0·3	...
,, ,, smooth, ,,	...	0·07 to 0·08	0·05	...	...	0·03
Bronze on lignum vitæ, ,,	...	...	...	...	0·05	...
Iron ,, ,, ,,	...	...	...	...	0·05	...
Bronze on bronze sliding, (S)	...	...	...	...	0·38	...
,, ,, ,, (Rn)	...	0·175	...	...	...	...
Steel ,, ,, ,,	...	0·139	...	...	...	...
Wrought iron on bronze sliding, ,,	...	0·135	...	...	...	...
Cast ,, ,, ,,	...	0·141	...	...	...	...
Steel on cast iron, ,,	...	0·151	...	...	...	...

CLXXII.—Co-efficients of Friction—*continued.*

Nature of the Surfaces.	Quite Dry.	Greasy Dry.	Well Lubricated with			Constant Flow of Oil.
			Oil.	Tallow or Soap.	Water.	
Steel on wrought iron, . . (Rn)	...	0·189	...	...	...	...
Wrought iron on cast iron, . ,,	...	0·170	...	...	...	...
,, ,, tin, . . ,,	...	0·181	...	...	...	...
,, ,, magnolia, (RHS)	...	0·060	...	...	...	...
,, ,, white metal, .	...	..	0·025	...	...	0·01 to 0·015
Loco axles, white metal, . (W)	...	...	0·017	...	...	...
Shafts on bronze, . . . (Rn)	...	...	...	0·028	...	...
Leather on metals, . . . (Rk)	0·56	0·23	0·15	...	0·36	...
Wrought-iron shaft on cast-iron bearing, speed 10 feet per minute,	..	...	0·079	...	...	...
Wrought iron shaft on cast-iron bearing, speed 60 feet per minute,	...	...	0·052	...	...	...
Wrought-iron shaft on cast-iron bearing, speed 100 feet per minute,	...	...	0·050	...	...	..
Load 50 lbs. p. sq. in., speed 50 ft., (G)	...	...	...	...	...	·0032
,, 50 ,, ,, ,, 110 ,, ,,	...	...	...	..	...	·0064
,, 50 ,, ,, ,, 190 ,, ,,	...	...	...	...	...	·0106
,, 150 ,, ,, ,, 50 ,, ,,	...	..	...	...	...	·0017
,, 150 ,, ,, ,, 110 ,, ,,	...	...	...	..	...	·0024
,, 150 ,, ,, ,, 190 ,, ,,	...	..	...	...	...	·0047
Speed of 10 ft. p. min., load 50 lbs., ,,	...	...	...	...	...	·0009
,, 10 ,, ,, 75 ,, ,,	...	...	...	...	...	·0007
,, 10 ,, ,, 150 ,, ,,	..	...	...	...	...	·0250
,, 15 ,, ,, 50 ,, ,,	...	...	..	...	...	·0012
,, 15 ,, ,, 150 ,, ,,	...	...	...	...	...	·0051
,, 7–8 ft. p. min., moderate load,	...	0·094	0·072	...	...	·064
,, ,, ,, heavy ,, ,,	...	0·051	0·046	...	...	·047
Railway axles and bearings, (Stahl)	...	...	0·100	0·067	...	...
Cast-iron steel tyres, . . (Gl)	0·36 to ·048	5 to 30 miles per hour.				
Steel tyres on steel rails, . ,,	0·11 to 0·04	10 to 50 ,, ,,				

Note.—Rk stands for Rankine; Rn, for Rennie; S, for Summers; W, for Wood; G, for Goodman; RHS, for Prof. Smith; Gl for Gallon and Westinghouse.

Table CLXXIII.—Conductivity of Sundry Materials.

	Relative Conductivity at 55° F.		Conducts Sound Feet per Second.
	Electric.	Thermal.	
Silver, annealed	100·0	100·0	...
Silver, hard drawn	92·1	...	...
Copper, annealed	94·0	91·8	...
Copper, hard drawn	91·3	...	11,300
Gold, hard drawn	71·8	53·2	...
Aluminium, annealed	51·6	31·3	...
Zinc, pressed	26·7	28·1	...
Brass tubes (70 cu. 30 zu.)	22·0	26·0	...
Platinum, annealed	16·6	8·4	...
Iron and mild steel, annealed	15·5	11·9	16,700
Nickel, annealed	12·05	14·0	...
Tin, pressed	7·65	8·5	...
German silver	6·00	6·0	...
Bismuth	2·00	1·8	...
Mercury, liquid	1·58	1·3	...

Table CLXXIV.—Various Gases, Properties of.

Name of Gas.	Density.	Molecular Weight.	No. of Lbs. per Cub. Ft.	No. of Cub. Ft. per Lb.	Air Required per Lb. Gas.	Temperature of Combustion.	Heat Generated by 1 Lb.	Specific Heat Vol. Const.
				Cub.Ft.	Lbs.	F.°	B.T.U.	
Hydrogen	1·0	2	·0056	178·83	34·79	4813	52,290	2·415
Oxygen	16·0	32	·0893	11·21	...	...	...	0·156
Steam	9·0	18	·0502	19·91	...	...	...	0·305
Nitrogen	14·0	28	·0785	12·76	...	...	...	0·174
Air	14·4	...	·0807	12·39	...	...	...	0·169
Carbon vapour	12·0	...	·0670	14·93	11·59	4988	20,461	...
,, monoxide	14·0	28	·0782	12·80	2·48	3494	4,383	0·173
,, bisulphide	38·0	76	·2124	4·71	5·48	6590	9,344	0·130
Benzine	39·0	78	·2080	4·81	13·38	5022	18,000	0·350
Acetylene	13·0	26	·0727	13·46	13·38	6120	21,856	...
Coal gas	4·7	...	·0320	31·60	13·89	4500	21,400	0·468
Blast furnace	14·0	...	·0790	12·65	1·00	2160	1,230	...
Ammonia	8·50	17	·0532	18·78	6·13	3760	9,670	0·391

Table CLXXV.—Pressure of Water due to various Heads.

Depth of Water	Pressure in Pounds per sq. in.	Pressure in Kilos. per sq. c/m.	Depth of Water.	Pressure in Pounds per sq. in.	Pressure in Kilos. per sq. c/m.	Depth of Water.	Pressure in Pounds per sq. in.	Pressure in Kilos. per sq. c/m.
1 in.	·03608	·002537	27 ft.	11·691	·82196	64 ft.	27·712	1·94836
2	·07216	·005074	28	12·124	·85240	65	28·145	1·97880
3	·10824	·007611	29	12·557	·88284	66	28·578	2·00925
4	·14432	·010148	30	12·990	·91329	67	29·011	2·03969
5	·18040	·012685	31	13·423	·94373	68	29·444	2·07013
6	·21648	·015222	32	13·856	·97417	69	29·877	2·10057
7	·25256	·017759	33	14·289	1·00462	70	30·310	2·13102
8	·28864	·020296	34	14·722	1·03406	71	30·743	2·16146
9	·32472	·022833	35	15·155	1·06450	72	31·176	2·19190
10	·36080	·025370	36	15·588	1·09495	73	31·609	2·22235
11	·39688	·027907	37	16·021	1·12539	74	32·042	2·25279
1 ft.	·433	·030443	38	16·454	1·15583	75	32·475	2·28323
2	·866	·060886	39	16·887	1·18627	76	32·908	2·31368
3	1·299	·091329	40	17·320	1·21773	77	33·341	2·34412
4	1·732	·121773	41	17·753	1·24817	78	33·774	2·37456
5	2·165	·152216	42	18·186	1·27861	79	34·207	2·40500
6	2·598	·182659	43	18·619	1·30906	80	34·640	2·43545
7	3·031	·213102	44	19·052	1·33950	81	35·073	2·46589
8	3·464	·243545	45	19·485	1·36994	82	35·506	2·49633
9	3·897	·273989	46	19·918	1·40039	83	35·939	2·52678
10	4·330	·30443	47	20·351	1·43083	84	36·372	2·55722
11	4·763	·33487	48	20·784	1·46127	85	36·805	2·58766
12	5·196	·36531	49	21·217	1·49171	86	37·238	2·61811
13	5·629	·39576	50	21·650	1·52216	87	37·671	2·64855
14	6·062	·42620	51	22·083	1·55260	88	38·104	2·67899
15	6·495	·45664	52	22·516	1·58304	89	38·537	2·70943
16	6·928	·48709	53	22·949	1·61349	90	38·970	2·73989
17	7·361	·51753	54	23·382	1·64393	91	39·403	2·77033
18	7·794	·54797	55	23·815	1·67437	92	39·836	2·80077
19	8·227	·57841	56	24·248	1·70482	93	40·269	2·83122
20	8·660	·60886	57	24·681	1·73526	94	40·702	2·86166
21	9·093	·63930	58	25·114	1·76570	95	41·135	2·89210
22	9·526	·66974	59	25·557	1·79614	96	41·568	2·92255
23	9·959	·70019	60	25·980	1·82659	97	42·001	2·95299
24	10·392	·73063	61	26·413	1·85703	98	42·434	2·98343
25	10·825	·76107	62	26·846	1·88747	99	42·867	3·01387
26	11·258	·79152	63	27·279	1·91792	100	43·300	3·04432

The above Table is calculated for fresh water at a temperature of 62° F

Table CLXXVI.—Coal Consumption in Tons per Day.

Indicated Horse-Power.	Pounds per Indicated Horse-Power per Hour.															
	1·0	1·1	1·2	1·3	1·4	1·5	1·6	1·7	1·8	1·9	2·0	2·1	2·2	2·3	2·4	2·5
10,000	107·1	117·8	128·5	139·3	150·0	160·7	171·4	182·1	192·8	203·5	214·2	225·0	235·7	246·4	257·1	267·8
9,000	96·4	106·0	115·7	125·3	135·0	144·6	154·2	163·9	173·5	183·2	192·7	202·5	212·1	221·7	231·4	241·0
8,000	85·7	94·2	102·8	111·4	120·0	128·5	137·1	145·7	154·2	162·4	171·4	180·0	188·5	197·1	205·7	214·2
7,000	75·0	82·5	90·0	97·5	105·0	112·5	120·0	127·5	135·0	142·5	150·0	157·5	165·0	172·5	180·0	187·5
6,000	64·3	70·7	77·1	83·5	90·0	96·4	102·8	109·2	115·7	122·1	128·5	135·0	141·4	147·8	154·2	160·7
5,000	53·5	58·9	64·2	69·6	75·0	80·3	85·7	91·0	96·4	101·8	107·1	112·5	117·8	123·2	128·5	134·0
4,000	42·8	47·1	51·4	55·7	60·0	64·2	68·5	72·8	77·1	81·4	85·7	90·0	94·2	98·5	102·8	107·1
3,000	32·1	35·3	38·6	41·7	45·0	48·2	51·4	54·6	57·9	61·0	64·2	67·5	70·7	73·9	77·1	80·3
2,000	21·4	23·5	25·7	27·8	30·0	32·1	34·2	36·4	38·5	41·7	42·8	45·0	47·1	49·2	51·4	53·5
1,000	10·7	11·8	12·8	13·9	15·0	16·1	17·1	18·2	19·3	20·3	21·4	22·5	23·6	24·6	25·7	26·8
900	9·64	10·62	11·57	12·53	13·50	14·46	15·42	16·39	17·35	18·32	19·27	20·25	21·21	22·17	23·14	24·10
800	8·57	9·42	10·28	11·14	12·00	12·85	13·71	14·57	15·42	16·24	17·14	18·00	18·85	19·71	20·57	21·42
700	7·50	8·25	9·00	9·75	10·50	11·25	12·00	12·75	13·50	14·25	15·00	15·75	16·50	17·25	18·00	18·75
600	6·43	7·07	7·71	8·35	9·00	9·64	10·28	10·92	11·57	12·21	12·85	13·50	14·14	14·78	15·42	16·07
500	5·35	5·89	6·42	6·96	7·50	8·03	8·57	9·10	9·64	10·18	10·71	11·25	11·78	12·32	12·85	13·40
400	4·28	4·71	5·14	5·57	6·00	6·42	6·85	7·28	7·71	8·14	8·57	9·00	9·42	9·85	10·28	10·71
300	3·21	3·53	3·86	4·17	4·50	4·82	5·14	5·46	5·78	6·10	6·42	6·75	7·07	7·39	7·71	8·03
200	2·14	2·35	2·57	2·78	3·00	3·21	3·42	3·64	3·85	4·07	4·28	4·50	4·71	4·92	5·14	5·35
100	1·07	1·18	1·28	1·39	1·50	1·61	1·71	1·82	1·92	2·03	2·14	2·25	2·35	2·46	2·57	2·68
90	·964	1·06	1·16	1·25	1·35	1·44	1·54	1·64	1·73	1·83	1·93	2·02	2·12	2·22	2·31	2·41
80	·857	·942	1·03	1·11	1·20	1·28	1·37	1·46	1·54	1·62	1·71	1·80	1·88	1·97	2·06	2·14
70	·750	·825	·900	·975	1·05	1·12	1·20	1·27	1·35	1·42	1·50	1·57	1·65	1·72	1·80	1·88
60	·643	·707	·771	·835	·900	·964	1·03	1·09	1·15	1·22	1·28	1·35	1·41	1·48	1·54	1·61
50	·535	·589	·642	·696	·750	·803	·857	·910	·964	1·01	1·07	1·12	1·17	1·23	1·29	1·34
40	·428	·471	·514	·557	·600	·642	·685	·728	·771	·814	·857	·900	·942	·985	1·03	1·07
30	·321	·353	·386	·417	·450	·482	·514	·546	·578	·610	·642	·675	·707	·739	·771	·803
20	·214	·235	·257	·278	·300	·321	·342	·364	·385	·407	·428	·450	·471	·492	·514	·536
10	·107	·118	·128	·139	·150	·161	·171	·182	·192	·203	·214	·225	·235	·246	·257	·268

Table CLXXVII.—Consumption of Oil and other Fuels in tons per 24 hours.

Horse Power.	Consumption per Horse-power hour in fractions of a pound.													
	0·25	0·30	0·35	0·40	0·45	0·50	0·55	0·60	0·65	0·70	0·75	0·80	0·85	0·90
100	0·269	0·323	0·376	0·430	0·484	0·538	0·591	0·645	0·699	0·753	0·806	0·860	0·914	0·968
150	0·403	0·484	0·564	0·645	0·726	0·806	0·887	0·968	1·048	1·129	1·209	1·290	1·371	1·451
200	0·538	0·645	0·753	0·860	0·967	1·075	1·183	1·290	1·398	1·505	1·613	1·720	1·828	1·935
250	0·672	0·806	0·941	1·075	1·209	1·344	1·478	1·613	1·747	1·881	2·016	2·150	2·284	2·419
300	0·806	0·968	1·129	1·290	1·451	1·613	1·774	1·935	2·096	2·258	2·418	2·580	2·741	2·903
350	0·941	1·129	1·317	1·505	1·693	1·881	2·069	2·258	2·446	2·634	2·821	3·070	3·198	3·386
400	1·075	1·290	1·505	1·720	1·935	2·150	2·365	2·580	2·795	3·010	3·224	3·440	3·655	3·870
450	1·210	1·451	1·693	1·935	2·177	2·419	2·661	2·903	3·144	3·386	3·628	3·870	4·112	4·354
500	1·344	1·613	1·881	2·150	2·419	2·688	2·956	3·225	3·494	3·763	4·031	4·300	4·569	4·838
600	1·613	1·935	2·258	2·580	2·902	3·225	3·547	3·870	4·193	4·515	4·837	5·160	5·483	5·805
700	1·882	2·258	2·634	3·010	3·386	3·763	4·139	4·515	4·891	5·268	5·643	6·020	6·396	6·773
800	2·150	2·580	3·010	3·440	3·870	4·300	4·730	5·160	5·590	6·020	6·449	6·880	7·310	7·740
900	2·419	2·901	3·386	3·870	4·354	4·838	5·321	5·805	6·289	6·773	7·255	7·740	8·224	8·708
1000	2·688	3·225	3·763	4·300	4·837	5·375	5·812	6·450	6·988	7·525	8·061	8·600	9·138	9·675
1250	3·360	4·031	4·703	5·375	6·047	6·719	7·266	8·063	8·734	9·406	10·08	10·75	11·42	12·09
1500	4·032	4·838	5·644	6·450	7·256	8·063	8·719	9·675	10·81	11·29	12·09	12·90	13·71	14·51
1750	4·704	5·644	6·584	7·525	8·465	9·406	10·17	11·29	12·23	13·17	14·11	15·05	15·99	16·93
2000	5·376	6·450	7·525	8·600	9·675	10·75	11·62	12·90	13·98	15·05	16·12	17·20	18·28	19·35
2500	6·720	8·063	9·406	10·75	12·09	13·44	14·53	16·13	17·47	18·81	20·15	21·50	22·84	24·19
3000	8·064	9·675	11·29	12·90	14·51	16·12	17·44	19·45	20·96	22·57	24·19	25·80	27·41	29·03
3500	9·408	11·29	13·17	15·05	16·93	18·81	20·34	22·68	24·46	26·33	28·22	30·10	31·98	33·87
4000	10·75	12·90	15·05	17·20	19·35	21·50	23·25	25·90	27·95	30·10	32·25	34·40	36·55	38·71
5000	13·44	16·13	18·81	21·50	24·19	26·87	29·16	32·22	34·94	37·61	40·30	43·00	45·69	48·39

Table CLXXVIII.—Comparison of Thermometers.

Fahrenheit degs.	Centigrade degs.	Reaumur degs.	Fahrenheit degs.	Centigrade degs.	Reaumur degs.	Fahrenheit degs.	Centigrade degs.	Reaumur degs.
−461·20	−274·00	−219·22	+	−	−	+	+	+
+	−	−	21·20	6·	4·80	44·60	7·	5·60
0·	17·78	14·22	22·	5·56	4·44	45·	7·23	5·77
0·50	17·50	14·	23·	5·	4·	45·50	7·50	6·
1·	17·23	13·77	24·	4·45	3·55	46·	7·78	6·22
1·40	17·	13·60	24·80	4·	3·20	46·40	8·	6·40
2·	16·67	13·33	25·	3·89	3·11	47·	8·34	6·66
2·75	16·25	13·	25·25	3·75	3·	47·75	8·75	7·
3·	16·11	12·88	26·	3·34	2·66	48·	8·89	7·11
3·20	16·	12·80	26·60	3·	2·40	48·20	9·	7·20
4·	15·56	12·44	27·	2·78	2·22	49·	9·45	7·55
5·	15·	12·	27·50	2·50	2·	50·	10·	8·
6·	14·45	11·55	28·	2·23	1·77	51·	10·56	8·44
6·80	14·	11·20	28·40	2·	1·60	51·80	11·	8·80
7·	13·89	11·11	29·	1·67	1·33	52·	11·11	8·88
7·25	13·75	11·	29·75	1·25	1·	52·25	11·25	9·
8·	13·34	10·66	30·	1·11	0·88	53·	11·67	9·33
8·60	13·	10·40	30·20	1·	0·80	53·60	12·	9·60
9·	12·78	10·22	31·	0·56	0·44	54·	12·23	9·77
9·50	12·50	10·	32·	0·	0·	54·50	12·50	10·
10·	12·23	9·77	+33·	+0·56	+0·44	55·	12·78	10·22
10·40	12·	9·60	33·80	1·	0·80	55·40	13·	10·40
11·	11·67	9·33	34·	1·11	0·88	56·	13·34	10·66
11·75	11·25	9·	34·25	1·25	1·	56·75	13·75	11·
12·	11·11	8·88	35·	1·67	1·33	57·	13·89	11·11
12·20	11·	8·80	35·60	2·	1·60	57·20	14·	11·20
13·	10·56	8·44	36·	2·23	1·77	58·	14·45	11·55
14·	10·	8·	36·50	2·50	2·	59·	15·	12·
15·	9·45	7·55	37·	2·78	2·22	60·	15·56	12·44
15·80	9·	7·20	37·40	3·	2·40	60·80	16·	12·80
16·	8·89	7·11	38·	3·34	2·66	61·	16·11	12·88
16·25	8·75	7·	38·75	3·75	3·	61·25	16·25	13·
17·	8·34	6·66	39·	3·89	3·11	62·	16·67	13·33
17·60	8·	6·40	39·20	4·	3·20	62·60	17·	13·60
18·	7·78	6·22	40·	4·45	3·55	63·	17·23	13·77
18·50	7·50	6·	41·	5·	4·	63·50	17·50	14·
19·	7·23	5·77	42·	5·56	4·44	64·	17·78	14·22
19·40	7·	5·60	42·80	6·	4·80	64·40	18·	14·40
20·	6·67	5·33	43·	6·11	4·88	65·	18·34	14·66
20·75	6·25	5·	43·25	6·25	5·	65·75	18·75	15·
21·	6·11	4·88	44·	6·67	5·33	66·	18·89	15·11

Note.—Temp. Fahr. = $\frac{9}{5}$ temp. Cent. + 32°.
Temp. Cent. = $\frac{5}{9}$ (temp Fahr. − 32°).

Table CLXXVIII.—Comparison of Thermometers—*continued.*

Fahrenheit degs.	Centigrade degs.	Reaumur degs.	Fahrenheit degs.	Centigrade degs.	Reaumur degs.	Fahrenheit degs.	Centigrade degs.	Reaumur degs.
+	+	+	+	+	+	+	+	+
66·20	19·	15·20	89·60	32·	25·60	111·20	44·	35·20
67·	19·45	15·55	90·	32·23	25·77	112·	44·45	35·55
68·	20·	16·	90·50	32·50	26·	113·	45·	36·
69·	20·56	16·44	91·	32·78	26·22	114·	45·56	36·44
69·80	21·	16·80	91·40	33·	26·40	114·80	46·	36·80
70·	21·11	16·88	92·	33·34	26·66	115·	46·11	36·88
70·25	21·25	17·	92·75	33·75	27·	115·25	46·25	37·
71·	21·67	17·33	93·	33·89	27·11	116·	46·67	37·33
71·60	22·	17·60	93·20	34·	27·20	116·60	47·	37·60
72·	22·23	17·77	94·	34·45	27·55	117·	47·23	37·77
72·50	22·50	18·	95·	35·	28·	117·50	47·50	38·
73·	22·78	18·22	96·	35·56	28·44	118·	47·78	38·22
73·40	23·	18·40	96·80	36·	28·80	118·40	48·	38·40
74·	23·34	18·66	97·	36·11	28·88	119·	48·34	38·66
74·75	23·75	19·	97·25	36·25	29·	119·75	48·75	39·
75·	23·89	19·11	98·	36·67	29·33	120·	48·89	39·11
75·20	24·	19·20	98·60	37·	29·60	120·20	49·	39·20
76·	24·45	19·55	99·	37·23	29·77	121·	49·45	39·55
77·	25·	20·	99·50	37·50	30·	122·	50·	40·
78·	25·56	20·44	100·	37·78	30·22	123·	50·56	40·44
78·80	26·	20·80	100·40	38·	30·40	123·80	51·	40·80
79·	26·11	20·88	101·	38·34	30·66	124·	51·11	40·88
79·25	26·25	21·	101·75	38·75	31·	124·25	51·25	41·
80·	26·67	21·33	102·	38·89	31·11	125·	51·67	41·33
80·60	27·	21·60	102·20	39·	31·20	125·60	52·	41·60
81·	27·23	21·77	103·	39·45	31·55	126	52·23	41·77
81·50	27·50	22·	104·	40·	32·	126·50	52·50	42·
82·	27·78	22·22	105·	40·56	32·44	127·	52·78	42·22
82·40	28·	22·40	105·80	41·	32·80	127·40	53·	42·40
83·	28·34	22·66	106·	41·11	32·88	128·	53·34	42·66
83·75	28·75	23·	106·25	41·25	33·	128·75	53·75	43·
84·	28·89	23·11	107·	41·67	33·33	129·	53·89	43·11
84·20	29·	23·20	107·60	42·	33·60	129·20	54·	43·20
85·	29·45	23·55	108·	42·23	33·77	130·	54·45	43·55
86·	30·	24·	108·50	42·50	34·	131·	55·	44·
87·	30·56	24·44	109·	42·78	34·22	132·	55·56	44·44
87·80	31·	24·80	109·40	43·	34·40	132·80	56·	44·80
88·	31·11	24·88	110·	43·34	34·66	133·	56·11	44·88
88·25	31·25	25·	110·75	43·75	35·	133·25	56·25	45·
89·	31·67	25·33	111·	43·89	35·11	134·	56·67	45·33

Table CLXXVIII.—Comparison of Thermometers—*continued.*

Fahrenheit degs.	Centigrade degs.	Reaumur degs.	Fahrenheit degs.	Centigrade degs.	Reaumur degs.	Fahrenheit degs.	Centigrade degs.	Reaumur degs.
+	+	+	+	+	+	+	+	+
134·60	57·	45·60	156·20	69·	55·20	179·60	82·	65·60
135·	57·23	45·77	157·	69·45	55·55	180·	82·23	65·77
135·50	57·50	46·	158·	70·	56·	180·50	82·50	66·
136·	57·78	46·22	159·	70·56	56·44	181·	82·78	66·22
136·40	58·	46·40	159·80	71·	56·80	181·40	83·	66·40
137·	58·34	46·66	160·	71·11	56·88	182·	83·34	66·66
137·75	58·75	47·	160·25	71·25	57·	182·75	83·75	67·
138·	58·89	47 11	161·	71·67	57·33	183·	83·89	67·11
138·20	59·	47·20	161·60	72·	57·60	183·20	84·	67·20
139·	59·45	47·55	162·	72·23	57·77	184·	84·45	67·55
140·	60·	48·	162·50	72·50	58·	185·	85·	68·
141·	60·56	48·44	163·	72·78	58·22	186·	85·56	68·44
141·80	61·	48·80	163·40	73·	58·40	186·80	86·	68·80
142·	61·11	48·88	164·	73·34	58·66	187·	86·11	68·88
142·25	61·25	49·	164·75	73·75	59·	187·25	86·25	69·
143·	61·67	49·33	165·	73·89	59·11	188·	86·67	69·33
143·60	62·	49·60	165·20	74·	59·20	188·60	87·	69·60
144·	62·23	49·77	166·	74·45	59·55	189·	87·23	69·77
144·50	62·50	50·	167·	75·	60·	189·50	87·50	70·
145·	62·78	50·22	168·	75·56	60·44	190·	87·78	70·22
145·40	63·	50·40	168·80	76·	60·80	190·40	88·	70·40
146·	63·34	50·66	169·	76·11	60·88	191·	88·34	70·66
146·75	63·75	51·	169·25	76·25	61·	191·75	88·75	71·
147·	63·89	51·11	170·	76·67	61·33	192·	88·89	71·11
147·20	64·	51·20	170·60	77·	61·60	192·20	89·	71·20
148·	64·45	51·55	171·	77·23	61·77	193·	89·45	71·55
149·	65·	52·	171·50	77·50	62·	194·	90·	72·
150·	65·56	52·44	172·	77·78	62·22	195·	90·56	72·44
150·80	66·	52·80	172·40	78·	62·40	195·80	91·	72·80
151·	66·11	52·88	173·	78·34	62·66	196·	91·11	72·88
151·25	66·25	53·	173·75	78·75	63·	196·25	91·25	73·
152·	66·67	53·33	174·	78·89	63·11	197·	91·67	73·33
152·60	67·	53·60	174·20	79·	63·20	197·60	92·	73·60
153·	67·23	53·77	175·	79·45	63·55	198·	92·23	73·77
153·50	67·50	54·	176·	80·	64·	198·50	92·50	74·
154·	67·78	54·22	177·	80·56	64·44	199·	92·78	74·22
154·40	68·	54·40	177·80	81·	64·80	199·40	93·	74·40
155·	68·34	54·66	178·	81·11	64·88	200·	93·34	74·66
155·75	68·75	55·	178·25	81·25	65·	200·75	93·75	75·
156·	68·89	55·11	179·	81·67	65·33	201·	93·89	75·11

Table CLXXVIII.—Comparison of Thermometers—*continued.*

Fahrenheit degs.	Centigrade degs.	Reaumur degs.	Fahrenheit degs.	Centigrade degs.	Reaumur degs.	Fahrenheit degs.	Centigrade degs.	Reaumur degs.
+	+	+	+	+	+	+	+	+
201·20	94·	75·20	224·60	107·	85·60	246·20	119·	95·20
202·	94·45	75·55	225·	107·23	85·77	247·	119·45	95·55
203·	95·	76·	225·50	107·50	86·	248·	120·	96·
204·	95·56	76·44	226·	107·78	86·22	249·	120·56	96·44
204·80	96·	76·80	226·40	108·	86·40	249·80	121·	96·80
205·	96·11	76·88	227·	108·34	86·66	250·	121·11	96·88
205·25	96·25	77·	227·75	108·75	87·	250·25	121·25	97·
206·	96·67	77·33	228·	108·89	87·11	251·	121·67	97·33
206·60	97·	77·60	228·20	109·	87·20	251·60	122·	97·60
207·	97·23	77·77	229·	109·45	87·55	252·	122·23	97·77
207·50	97·50	78·	230·	110·	88·	252·50	122·50	98·
208·	97·78	78·22	231·	110·56	88·44	253·	122·78	98·22
208·40	98·	78·40	231·80	111·	88·80	253·40	123·	98·40
209·	98·34	78·66	232·	111·11	88·88	254·	123·34	98·66
209·75	98·75	79·	232·25	111·25	89·	254·75	123·75	99·
210·	98·89	79·11	233·	111·67	89·33	255·	123·89	99·11
210·20	99·	79·20	233·60	112·	89·60	255·20	124·	99·20
211·	99·45	79·55	234·	112·23	89·77	256·	124·45	99·55
212·	100·	80·	234·50	112·50	90·	257·	125·	100·
213·	100·56	80·44	235·	112·78	90·22	258·	125·56	100·44
213·80	101·	80·80	235·40	113·	90·40	258·80	126·	100·80
214·	101·11	80·88	236·	113·34	90·66	259·	126·11	100·88
214·25	101·25	81·	236·75	113·75	91·	259·25	126·25	101·
215·	101·67	81·33	237·	113·89	91·11	260·	126·67	101·33
215·60	102·	81·60	237·20	114·	91·20	260·60	127·	101·60
216·	102·23	81·77	238·	114·45	91·55	261·	127·23	101·77
216·50	102·50	82·	239·	115·	92·	261·50	127·50	102·
217·	102·78	82·22	240·	115·56	92·44	262·	127·78	102·22
217·40	103·	82·40	240·80	116·	92·80	262·40	128·	102·40
218·	103·34	82·66	241·	116·11	92·88	263·	128·34	102·66
218·75	103·75	83·	241·25	116·25	93·	263·75	128·75	103·
219·	103·89	83·11	242·	116·67	93·33	264·	128·89	103·11
219·20	104·	83·20	242·60	117·	93·60	264·20	129·	103·20
220·	104·45	83·55	243·	117·23	93·77	265·	129·45	103·55
221·	105·	84·	243·50	117·50	94·	266·	130·	104·
222·	105·56	84·44	244·	117·78	94·22	267·	130·26	104·44
222·80	106·	84·80	244·40	118·	94·40	267·80	131·	104·80
223·	106·11	84·88	245·	118·34	94·66	268·	131·11	104·88
223·25	106·25	85·	245·75	118·75	95·	268·25	131·25	105·
224·	106·67	85·33	246·	118·89	95·11	269·	131·67	105·33

Table CLXXVIII.—Comparison of Thermometers—*continued.*

Fahrenheit degs.	Centigrade degs.	Reaumur degs.	Fahrenheit degs.	Centigrade degs.	Reaumur degs.	Fahrenheit degs.	Centigrade degs.	Reaumur degs.
+	+	+	+	+	+	+	+	+
269·60	132·	105·60	291·20	144·	115·20	314·60	157·	125·60
270·	132·23	105·77	292·	144·45	115·55	315·	157·23	125·77
270·50	132·50	106·	293·	145·	116·	315·50	157·50	126·
271·	132·78	106·22	294·	145·56	116·44	316·	157·78	126·22
271·40	133·	106·40	294·80	146·	116·80	316·40	158·	126·40
272·	133·34	106·66	295·	146·11	116·88	317·	158·34	126·66
272·75	133·75	107·	295·25	146·25	117·	317·75	158·75	127·
273·	133·89	107·11	296·	146·67	117·33	318·	158·89	127·11
273·20	134·	107·20	296·60	147·	117·60	318·20	159·	127·20
274·	134·45	107·55	297·	147·23	117·77	319·	159·45	127·55
275·	135·	108·	297·50	147·50	118·	320·	160·	128·
276·	135·56	108·44	298·	147·78	118·22	321·	160·56	128·44
276·80	136·	108·80	298·40	148·	118·40	321·80	161·	128·80
277·	136·11	108·88	299·	148·34	118·66	322·	161·11	128·88
277·25	136·25	109·	299·75	148·75	119·	322·25	161·25	129·
278·	136·67	109·33	300·	148·89	119·11	323·	161·67	129·33
278·60	137·	109·60	300·20	149·	119·20	323·60	162·	129·60
279·	137·23	109·77	301·	149·45	119·55	324·	162·23	129·77
279·50	137·50	110·	302·	150·	120·	324·50	162·50	130·
280·	137·78	110·22	303·	150·56	120·44	325·	162·78	130·22
280·40	138·	110·40	303·80	151·	120·80	325·40	163·	130·40
281·	138·34	110·66	304·	151·11	120·88	326·	163·34	130·66
281·75	138·75	111·	304·25	151·25	121·	326·75	163·75	131·
282·	138·89	111·11	305·	151·67	121·33	327·	163·89	131·11
282·20	139·	111·20	305·60	152·	121·60	327·20	164·	131·20
283·	139·45	111·55	306·	152·23	121·77	328·	164·45	131·55
284·	140·	112·	306·50	152·50	122·	329·	165·	132·
285·	140·56	112·44	307·	152·78	122·22	330·	165·56	132·44
285·80	141·	112·80	307·40	153·	122·40	330·80	166·	132·80
286·	141·11	112·88	308·	153·34	122·66	331·	166·11	132·88
286·25	141·25	113·	308·75	153·75	123·	331·25	166·25	133·
287·	141·67	113·33	309·	153·89	123·11	332·	166·67	133·33
287·60	142·	113·60	309·20	154·	123·20	332·60	167·	133·60
288·	142·23	113·77	310·	154·45	123·55	333·	167·23	133·77
288·50	142·50	114·	311·	155·	124·	333·50	167·50	134·
289·	142·78	114·22	312·	155·56	124·44	334·	167·78	134·22
289·40	143·	114·40	312·80	156·	124·80	334·40	168·	134·40
290·	143·34	114·66	313·	156·11	124·88	335·	168·34	134·66
290·75	143·75	115·	313·25	156·25	125·	335·75	168·75	135·
291·	143·89	115·11	314·	156·67	125·33	336·	168·89	135·11

Table CLXXVIII.—Comparison of Thermometers—*continued.*

Fahrenheit degs.	Centigrade degs.	Reaumur degs.	Fahrenheit degs.	Centigrade degs.	Reaumur degs.	Fahrenheit degs.	Centigrade degs.	Reaumur degs.
+	+	+	+	+	+	+	+	+
336·20	169·	135·20	359·60	182·	145·60	381·20	194·	155·20
337·	169·45	135·55	360·	182·23	145·77	382·	194·45	155·55
338·	170·	136·	360·50	182·50	146·	383·	195·	156·
339·	170·56	136·44	361·	182·78	146·22	384·	195·56	156·44
339·80	171·	136·80	361·40	183·	146·40	384·80	196·	156·80
340·	171·11	136·88	362·	183·34	146·66	385·	196·11	156·88
340·25	171·25	137·	362·75	183·75	147·	385·25	196·25	157·
341·	171·67	137·33	363·	183·89	147·11	386·	196·67	157·33
341·60	172·	137·60	363·20	184·	147·20	386·60	197·	157·60
342·	172·23	137·77	364·	184·45	147·55	387·	197·23	157·77
342·50	172·50	138·	365·	185·	148·	387·50	197·50	158·
343·	172·78	138·22	366·	185·56	148·44	388·	197·78	158·22
343·40	173·	138·40	366·80	186·	148·80	388·40	198·	158·40
344·	173·34	138·66	367·	186·11	148·88	389·	198·34	158·66
344·75	173·75	139·	367·25	186·25	149·	389·75	198·75	159·
345·	173·89	139·11	368·	186·67	149·33	390·	198·89	159·11
345·20	174·	139·20	368·60	187·	149·60	390·20	199·	159·20
346·	174·45	139·55	369·	187·23	149·77	391·	199·45	159·55
347·	175·	140·	369·50	187·50	150·	392·	200·	160·
348·	175·56	140·44	370·	187·78	150·22	393·	200·56	160·44
348·80	176·	140·80	370·40	188·	150·40	393·80	201·	160·80
349·	176·11	140·88	371·	188·34	150·66	394·	201·11	160·88
349·25	176·25	141·	371·75	188·75	151·	394·25	201·25	161·
350·	176·67	141·33	372·	188·89	151·11	395·	201·67	161·33
350·60	177·	141·60	372·20	189·	151·20	395·60	202·	161·60
351·	177·23	141·77	373·	189·45	151·55	396·	202·23	161·77
351·50	177·50	142·	374·	190·	152·	396·50	202·50	162·
352·	177·78	142·22	375·	190·56	152·44	397·	202·78	162·22
352·40	178·	142·40	375·80	191·	152·80	397·40	203·	162·40
353·	178·34	142·66	376·	191·11	152·88	398·	203·34	162·66
353·75	178·75	143·	376·25	191·25	153·	398·75	203·75	163·
354·	178·89	143·11	377·	191·67	153·33	399·	203·89	163·11
354·20	179·	143·20	377·60	192·	153·60	399·20	204·	163·20
355·	179·45	143·55	378·	192·23	153·77	400·	204·45	163·55
356·	180·	144·	378·50	192·50	154·	401·	205·	164·
357·	180·56	144·44	379·	192·78	154·22	402·	205·56	164·44
357·80	181·	144·80	379·40	193·	154·40	402·80	206·	164·80
358·	181·11	144·88	380·	193·34	154·66	403·	206·11	164·88
358·25	181·25	145·	380·75	193·75	155·	403·25	206·25	165·
359·	181·67	145·33	381·	193·89	155·11	404·	206·67	165·33

Table CLXXVIII.—Comparison of Thermometers—*continued.*

Fahrenheit degs.	Centigrade degs.	Reaumur degs.	Fahrenheit degs.	Centigrade degs.	Reaumur degs.	Fahrenheit degs.	Centigrade degs.	Reaumur degs.
+	+	+	+	+	+	+	+	+
404·60	207·	165·60	426·	218·89	175·11	448·25	231·25	185·
405·	207·23	165·77	426·20	219·	175·20	449·	231·67	185·33
405·50	207·50	166·	427·	219·45	175·55	449·60	232·	185·60
406·	208·75	163·22	428·	220·	176·	450·	232·23	185·77
406·40	208·	166·40	429·	220·56	176·44	455·	235·	188·
407·	208·34	166·66	429·80	221·	176·80	464·	240·	192·
407·75	208·75	167·	430·	221·11	176·88	473·	245·	196·
408·	208·89	167·11	430·25	221·25	177·	482·	250·	200·
408·20	209·	167·20	431·	221·67	177·33	491·	255·	204·
409·	209·45	167·55	431·60	222·	177·60	500·	260·	208·
410·	210·	168·	432·	222·23	177·77	509·	265·	212·
411·	210·56	168·44	432·50	222·50	178·	518·	270·	216·
411·80	211·	168·80	433·	222·78	178·22	527·	275·	220·
412·	211·11	168·88	433·40	223·	178·40	536·	280·	224·
412·25	211·25	169·	434·	223·34	178·66	545·	285·	228·
413·	211·67	169·33	434·75	223·75	179·	554·	290·	232·
413·60	212·	169·60	435·	223·89	179·11	563·	295·	236·
414·	212·23	169·77	435·20	224·	179·20	572·	300·	240·
414·50	212·50	170·	436·	224·45	179·55	581·	305·	244·
415·	212·78	170·22	437·	225·	180·	590·	310·	248·
415·40	213·	170·40	438·	225·56	180·44	599·	315·	252·
416·	213·34	170·66	438·80	226·	180·80	608·	320·	256·
416·75	213·75	171·	439·	226·11	180·88	617·	325·	260·
417·	213·89	171·11	439·25	226·25	181·	626·	330·	264·
417·20	214·	171·20	440·	226·67	181·33	635·	335·	268·
418·	214·45	171·55	440·60	227·	181·60	644·	340·	272·
419·	215·	172·	441·	227·23	181·77	653·	345·	276·
420·	215·56	172·44	441·50	227·50	182·	662·	350·	280·
420·80	216·	172·80	442·	227·78	182·22	671·	355·	284·
421·	216·11	172·88	442·40	228·	182·40	680·	360·	288·
421·25	216·25	173·	443·	228·34	182·66	689·	365·	292·
422·	216·67	173·33	443·75	228·75	183·	698·	370·	296·
422·60	217·	173·60	444·	228·89	183·11	707·	375·	300·
423·	217·23	173·77	444·20	229·	183·20	716·	380·	304·
423·50	217·50	174·	445·	229·45	183·55	725·	385·	308·
424·	217·78	174·22	446·	230·	184·	734·	390·	312·
424·40	218·	174·40	447·	230·56	184·44	743·	395·	316·
425·	218·34	174·66	447·80	231·	184·80	752·	400·	320·
425·75	218·75	175·	448·	231·11	184·88			

Table CLXXIX.—Properties of Saturated Steam.

Pressure per Square Inch from Mean Atmospheric Pressure. lbs.	Temperature in Fahrenheit Degrees.	Specific or Relative Volume of the Steam.	Density or Weight of 1 Cubic Foot of the Steam. lb.	Cubic Feet of the Steam per lb.	Latent Heat of Evaporation in Thermal Units per lb. of the Steam.	Total Heat in Thermal Units from 32° Fahrenheit per lb. of the Steam.	Absolute Pressure per Square Inch. lbs.
−14	90·4	28740	·002170	460·7	1051·1	1109·5	·7
−13	120·3	12480	·004998	200·1	1030·3	1118·6	1·7
−12	137·5	8080	·007720	129·5	1018·3	1123·9	2·7
−11	149·8	6009	·01038	96·32	1009·7	1127·6	3·7
−10	159·7	4799	·01300	76·92	1002·8	1130·7	4·7
−9	167·9	4003	·01558	64·17	997·0	1133·2	5·7
−8	174·9	3439	·01814	55·12	992·1	1135·3	6·7
−7	181·1	3010	·02072	48·25	987·8	1137·2	7·7
−6	186·7	2690	·02319	43·12	983·9	1138·9	8·7
−5	191·8	2428	·02568	38·93	980·3	1140·4	9·7
−4	196·4	2215	·02817	35·50	977·1	1141·9	10·7
−3	200·7	2036	·03063	32·64	974·0	1143·2	11·7
−2	204·7	1886	·03307	30·24	971·2	1144·4	12·7
−1	208·5	1755	·03553	28·14	968·5	1145·5	13·7
Atmosphere.	212·0	1643	·03797	26·34	966·1	1146·6	14·7
1	215·3	1544	·04039	24·76	963·8	1147·6	15·7
2	218·5	1457	·04280	23·36	961·5	1148·6	16·7
3	221·5	1380	·04521	22·12	959·4	1149·5	17·7
4	224·4	1310	·04761	21·0	957·3	1150·4	18·7
5	227·1	1248	·05000	20·0	955·4	1151·2	19·7
6	229·8	1191	·05238	19·09	953·5	1152·0	20·7
7	232·3	1139	·05475	18·26	951·8	1152·8	21·7
8	234·7	1092	·05712	17·51	950·1	1153·5	22·7
9	237·1	1049	·05949	16·81	948·4	1154·3	23·7
10	239·4	1009	·06184	16·17	946·7	1155·0	24·7
11	241·6	971·8	·06419	15·58	945·2	1155·6	25·7
12	243·7	937·5	·06654	15·03	943·7	1156·3	26·7
13	245·8	905·7	·06888	14·52	942·2	1156·9	27·7
14	247·8	876·0	·07122	14·04	940·8	1157·5	28·7
15	249·7	818·2	·07355	13·60	939·4	1158·1	29·7
16	251·6	822·2	·07587	13·18	938·1	1158·7	30·7

Table CLXXIX.—Properties of Saturated Steam—*continued.*

Pressure per Square Inch from Mean Atmospheric Pressure. lbs.	Temperature in Fahrenheit Degrees.	Specific or Relative Volume of the Steam.	Density or Weight of 1 Cubic Foot of the Steam. lb.	Cubic Feet of the Steam per lb.	Latent Heat of Evaporation in Thermal Units per lb. of the Steam.	Total Heat in Thermal Units from 32° Fahrenheit per lb. of the Steam.	Absolute Pressure per Square Inch. lbs.
17	253·5	797·8	·07819	12·79	936·8	1159·3	31·7
18	255·3	774·9	·08050	12·42	935·5	1159·8	32·7
19	257·0	753·2	·08282	12·07	934·3	1160·3	33·7
20	258·7	732·8	·08513	11·75	933·1	1160·9	34·7
21	260·4	713·5	·08743	11·44	931·9	1161·4	35·7
22	262·0	695·2	·08973	11·14	930·7	1161·9	36·7
23	263·6	677·9	·09203	10·87	929·6	1162·3	37·7
24	265·2	661·4	·09433	10·60	928·4	1162·8	38·7
25	266·7	645·7	·09661	10·35	927·4	1163·3	39·7
26	268·2	630·8	·09890	10·11	926·3	1163·8	40·7
27	269·7	616·6	·1011	9·883	925·2	1164·2	41·7
28	271·1	603·0	·1034	9·666	924·3	1164·6	42·7
29	272·6	590·0	·1057	9·458	923·2	1165·1	43·7
30	273·9	577·6	·1080	9·259	922·3	1165·5	44·7
31	275·3	565·7	·1102	9·068	921·3	1165·9	45·7
32	276·7	554·3	·1125	8·885	920·3	1166·3	46·7
33	278·0	543·4	·1148	8·710	919·3	1166·7	47·7
34	279·3	532·9	·1170	8·541	918·4	1167·1	48·7
35	280·5	522·8	·1193	8·380	917·6	1167·5	49·7
36	281·8	513·1	·1215	8·225	916·6	1167·9	50·7
37	283·0	503·8	·1238	8·075	915·8	1168·3	51·7
38	284·2	494·8	·1260	7·931	914·9	1168·6	52·7
39	285·4	486·1	·1283	7·792	914·1	1169·0	53·7
40	286·6	477 7	·1305	7·658	913·2	1169·4	54·7
41	287·8	469·7	·1328	7·529	912·3	1169·7	55·7
42	288·9	461·9	·1350	7·404	911·6	1170·1	56·7
43	290·1	454·4	·1373	7·283	910·7	1170·4	57·7
44	291·2	447·1	·1395	7·167	909·9	1170·8	58·7
45	292·3	440·0	·1417	7·054	909·1	1171·1	59·7
46	293·4	433·2	·1440	6·944	908·3	1171·4	60·7
47	294·4	426·6	·1462	6·839	907·6	1171·7	61·7

Table CLXXIX.—Properties of Saturated Steam—*continued.*

Pressure per Square Inch from Mean Atmospheric Pressure. lbs.	Temperature in Fahrenheit Degrees.	Specific or Relative Volume of the Steam.	Density or Weight of 1 Cubic Foot of the Steam. lb.	Cubic Feet of the Steam per lb.	Latent Heat of Evaporation in Thermal Units per lb. of the Steam.	Total Heat in Thermal Units from 32° Fahrenheit per lb. of the Steam.	Absolute Pressure per Square inch. lbs.
48	295·5	420·2	·1484	6·736	906·8	1172·1	62·7
49	296·5	414·1	·1506	6·636	906·1	1172·4	63·7
50	297·5	408·0	·1529	6·540	905·4	1172·7	64·7
51	298·6	402·2	·1551	6·446	904·6	1173·0	65·7
52	299·6	396·5	·1573	6·356	903·9	1173·3	66·7
53	300·6	391·0	·1595	6·267	903·2	1173·6	67·7
54	301·5	385·6	·1617	6·181	902·5	1173·9	68·7
55	302·5	380·4	·1639	6·098	901·8	1174·2	69·7
56	303·5	375·4	·1661	6·017	901·1	1174·5	70·7
57	304·4	370·4	·1684	5·938	900·5	1174·8	71·7
58	305·3	365·6	·1706	5·861	899·8	1175·1	72·7
59	306·3	361·0	·1728	5·786	899·1	1175·4	73·7
60	307·2	356·4	·1750	5·714	898·5	1175·6	74·7
61	308·1	352·0	·1772	5·643	897·8	1175·9	75·7
62	309·0	347·7	·1794	5·573	897·2	1176·2	76·7
63	309·9	343·5	·1816	5·506	896·5	1176·5	77·7
64	310·8	339·4	·1838	5·440	895·9	1176·7	78·7
65	311·6	335·4	·1860	5·376	895·3	1177·0	79·7
66	312·5	331·5	·1882	5·313	894·7	1177·3	80·7
67	313·3	327·7	·1904	5·252	894·1	1177·5	81·7
68	314·2	323·9	·1926	5·192	893·4	1177·8	82·7
69	315·0	320·3	·1947	5·134	89z·9	1178·0	83·7
70	315·8	316·7	·1969	5·077	892·2	1178·3	84·7
71	316·7	313·3	·1991	5·021	891·6	1178·5	85·7
72	317·5	309·9	·2013	4·967	891·1	1178·8	86·7
73	318·3	306·5	·2035	4·914	890·5	1179·0	87·7
74	319·1	303·3	·2056	4·862	889·9	1179·3	88·7
75	319·9	300·1	·2078	4·811	889·3	1179·5	89·7
76	320·7	297·0	·2100	4·761	888·8	1179·8	90·7
77	321·4	294·0	·2122	4·712	888·3	1180·0	91·7
78	322·2	291·0	·2144	4·664	887·7	1180·2	92·7

Table CLXXIX.—Properties of Saturated Steam—*continued.*

Pressure per Square Inch from Mean Atmospheric Pressure. lbs.	Temperature in Fahrenheit Degrees.	Specific or Relative Volume of the Steam.	Density or Weight of 1 Cubic Foot of the Steam. lb.	Cubic Feet of the Steam per lb.	Latent Heat of Evaporation in Thermal Units per lb. of the Steam.	Total Heat in Thermal Units from 32° Fahrenheit per lb. of the Steam.	Absolute Pressure per Square Inch. lbs.
79	323·0	288·0	·2165	4·617	887·1	1180·5	93·7
80	323·8	285·2	·2187	4·572	886·5	1180 7	94·7
81	324·5	282·4	·2209	4·527	886·0	1180·9	95·7
82	325·2	279·6	·2230	4·483	885·5	1181·1	96·7
83	326·0	276·9	·2252	4·439	885·0	1181·4	97·7
84	326·7	274·3	·2274	4·397	884·4	1181·6	98·7
85	327·4	271·7	·2295	4·356	883·9	1181·8	99·7
86	328·1	269·2	·2317	4·315	883·4	1182·0	100·7
87	328·9	266·7	·2339	4·275	882·9	1182·3	101·7
88	329·6	264·3	·2360	4·236	882·4	1182·5	102·7
89	330·3	261·9	·2382	4·198	881·9	1182·7	103·7
90	331·0	259·5	·2404	4·160	881·3	1182·9	104·7
91	331·7	257·2	·2425	4·123	880·8	1183·1	105·7
92	332·3	254·9	·2447	4·087	880·4	1183·3	106·7
93	333·0	252·5	·2468	4·051	879·9	1183·5	107·7
94	333·7	250·5	·2490	4·016	879·4	1183·7	108·7
95	334·4	248·4	·2511	3·981	878·9	1183·9	109·7
96	335·1	246·3	·2533	3·948	878·4	1184·2	110·7
97	335·7	244·2	·2554	3·914	878·0	1184·3	111·7
98	336·4	242·2	·2576	3·882	877·4	1184·6	112·7
99	337·0	240·1	·2598	3·849	877·0	1184·7	113·7
100	337·7	238·2	·2619	3·818	876·5	1184·9	114·7
101	338·3	236·2	·2640	3·787	876·1	1185·1	115·7
102	339·0	234·3	·2662	3·757	875·6	1185·3	116·7
103	339·6	232·5	·2683	3·727	875·1	1185·5	117·7
104	340·2	230·6	·2704	3·697	874·7	1185·7	118·7
105	340·9	228·8	·2726	3·668	874·2	1185·9	119·7
106	341·5	227·0	·2747	3·639	873·8	1186·1	120·7
107	342·1	225·1	·2769	3·611	873·3	1186·3	121·7
108	342·7	223·6	·2790	3·584	872·9	1186·5	122·7
109	343·3	221·9	·2811	3·556	872·5	1186·7	123·7

Table CLXXIX.—Properties of Saturated Steam—*continued.*

Pressure per Square Inch from Mean Atmospheric Pressure. lbs.	Temperature in Fahrenheit Degrees.	Specific or Relative Volume of the Steam.	Density or Weight of 1 Cubic Foot of the Steam. lb.	Cubic Feet of the Steam per lb.	Latent Heat of Evaporation in Thermal Units per lb. of the Steam.	Total Heat in Thermal Units from 32° Fahrenheit per lb. of the Steam.	Absolute Pressure per Square Inch. lbs.
110	343·9	220·1	·2833	3·530	872·0	1186·8	124·7
111	344·5	218·5	·2854	3·503	871·6	1187·0	125·7
112	345·1	216·9	·2875	3·477	871·2	1187·2	126·7
113	345·7	215·3	·2897	3·452	870·7	1187·4	127·7
114	346·3	213·7	·2918	3·426	870·3	1187·6	128·7
115	346·9	212·2	·2939	3·401	869·9	1187·8	129·7
116	347·5	210·7	·2961	3·377	869·4	1187·9	130·7
117	348·1	209·2	·2982	3·353	869·0	1188·1	131·7
118	348·7	207·7	·3003	3·329	868·6	1188·3	132·7
119	349·2	206·2	·3025	3·306	868·2	1188·5	133·7
120	349·8	204·8	·3046	3·283	867·8	1188·6	134·7
121	350·4	203·4	·3067	3·260	867·3	1188·8	135·7
122	351·0	202·0	·3088	3·237	866·9	1189·0	136·7
123	351·5	200·6	·3110	3·215	866·5	1189·2	137·7
124	352·1	199·2	·3131	3·194	866·1	1189·3	138·7
125	352·6	197·9	·3152	3·172	865·7	1189·5	139·7
126	353·2	196·6	·3173	3·151	865·3	1189·7	140·7
127	353·7	195·3	·3194	3·130	864·9	1189·8	141·7
128	354·3	194·0	·3216	3·109	864·5	1190·0	142·7
129	354·8	192·7	·3237	3·089	864·1	1190·2	143·7
130	355·4	191·5	·3258	3·069	863·7	1190·3	144·7
131	355·9	190·2	·3279	3·049	863·3	1190·5	145·7
132	356·4	189·0	·3300	3·030	863·0	1190·7	146·7
133	357·0	187·8	·3321	3·010	862·5	1190·8	147·7
134	357·5	186·6	·3343	2·991	862·2	1191·0	148·7
135	358·0	185·4	·3364	2·973	861·8	1191·1	149·7
136	358·5	184·3	·3385	2·954	861·4	1191·3	150·7
137	359·1	182·9	·3406	2·936	861·0	1191·5	151·7
138	359·6	182·0	·3427	2·918	860·6	1191·6	152·7
139	360·1	180·9	·3448	2·900	860·3	1191·8	153·7
140	360·6	179·8	·3469	2·882	859·9	1191·9	154·7

Table CLXXIX.—Properties of Saturated Steam—*continued.*

Pressure per Square Inch from Mean Atmospheric Pressure. lbs.	Temperature in Fahrenheit Degrees.	Specific or Relative Volume of the Steam.	Density or Weight of 1 Cubic Foot of the Steam. lb.	Cubic Feet of the Steam per lb.	Latent Heat of Evaporation in Thermal Units per lb. of the Steam.	Total Heat in Thermal Units from 32° Fahrenheit per lb. of the Steam.	Absolute Pressure per Square Inch. lbs.
141	361·1	178·7	·3490	2·865	859·6	1192·1	155·7
142	361·6	177·6	·3511	2·848	859·2	1192·2	156·7
143	362·1	176·6	·3532	2·831	858·8	1192·4	157·7
144	362·6	175·5	·3554	2·814	858·5	1192·5	158·7
145	363·1	174·5	·3575	2·797	858·1	1192·7	159·7
146	363·6	173·5	·3596	2·781	857·7	1192·8	160·7
147	364·1	172·5	·3617	2·765	857·4	1193·0	161·7
148	364·6	171·5	·3638	2·749	857·0	1193·2	162·7
149	365·1	170·5	·3659	2·733	856·6	1193·3	163·7
150	365·6	169·5	·3680	2·717	856·3	1193·5	164·7
151	366·1	168·6	·3701	2·702	855·9	1193·6	165·7
152	366·6	167·6	·3722	2·687	855·5	1193·8	166·7
153	367·1	166·7	·3743	2·672	855·2	1193·9	167·7
154	367·5	165·7	·3764	2·657	854·9	1194·0	168·7
155	368·0	164·8	·3785	2·642	854·5	1194·2	169·7
156	368·5	163·9	·3806	2·627	854·2	1194·3	170·7
157	369·0	163·0	·3827	2·613	853·8	1194·5	171·7
158	369·4	162·1	·3847	2·599	853·5	1194·6	172·7
159	369·9	161·2	·3868	2·585	853·1	1194·8	173·7
160	370·4	160·4	·3889	2·571	852·8	1194·9	174·7
161	370·8	159·5	·3910	2·557	852·5	1195·0	175·7
162	371·3	158·7	·3931	2·543	852·1	1195·2	176·7
163	371·7	157·8	·3952	2·530	851·8	1195·3	177·7
164	372·2	157·0	·3973	2·517	851·5	1195·5	178·7
165	372·7	156·2	·3994	2·504	851·1	1195·6	179·7
166	373·1	155·4	·4015	2·491	850·8	1195·7	180·7
167	373·6	154·6	·4036	2·478	850·4	1195·9	181·7
168	374·0	153·8	·4057	2·465	850·2	1196·0	182·7
169	374·5	153·0	·4077	2·452	849·8	1196·2	183·7
170	374·9	152·2	·4098	2·440	849·5	1196·3	184·7
171	375·4	151·4	·4119	2·427	849·1	1196·4	185·7

Table CLXXIX.—Properties of Saturated Steam—*continued.*

Pressure per Square Inch from Mean Atmospheric Pressure. lbs.	Temperature in Fahrenheit Degrees.	Specific or Relative Volume of the Steam.	Density or Weight of 1 Cubic Foot of the Steam. lb.	Cubic Feet of the Steam per lb.	Latent Heat of Evaporation in Thermal Units per lb. of the Steam.	Total Heat in Thermal Units from 32° Fahrenheit per lb. of the Steam.	Absolute Pressure per Square Inch. lbs.
172	375·8	150·7	·4140	2·415	848·8	1196·6	186·7
173	376·2	149·9	·4161	2·403	848·5	1196·7	187·7
174	376·7	149·2	·4182	2·391	848·2	1196·8	188·7
175	377·1	148·4	·4203	2·379	847·9	1197·0	189·7
176	377·5	147·7	·4223	2·368	847·6	1197·1	190·7
177	378·0	147·0	·4244	2·356	847·2	1197·2	191·7
178	378·4	146·3	·4265	2·344	846·9	1197·4	192·7
179	378·8	145·5	·4286	2·333	846·7	1197·5	193·7
180	379·3	144·8	·4307	2·322	846·3	1197·6	194·7
181	379·7	144·1	·4327	2·311	846·0	1197·8	195·7
182	380·1	143·5	·4348	2·300	845·7	1197·9	196·7
183	380·5	142·8	·4369	2·289	845·4	1198·0	197·7
184	381·0	142·1	·4390	2·278	845·0	1198·2	198·7
185	381·4	141·4	·4410	2·267	844·8	1198·3	199·7
186	381·8	140·8	·4431	2·257	844·5	1198·4	200·7
187	382·2	140·1	·4452	2·246	844·2	1198·5	201·7
188	382·6	139·5	·4473	2·236	843·9	1198·6	202·7
189	383·0	138·8	·4493	2·225	843·6	1198·8	203·7
190	383·5	138·2	·4514	2·215	843·2	1198·8	204·7
191	383·9	137·5	·4535	2·205	842·9	1199·0	205·7
192	384·3	136·9	·4556	2·195	842·6	1199·2	206·7
193	384·7	136·3	·4576	2·185	842·3	1199·3	207·7
194	385·1	135·7	·4597	2·175	842·0	1199·4	208·7
195	385·5	135·1	·4618	2·165	841·8	1199·5	209·7
196	385·9	134·5	·4639	2·156	841·5	1199·6	210·7
197	386·3	133·9	·4659	2·146	841·2	1199·8	211·7
198	386·7	133·3	·4680	2·137	840·9	1199·9	212·7
199	387·1	132·7	·4701	2·127	840·6	1200·0	213·7
200	387·5	132·1	·4721	2·118	840·3	1200·1	214·7

Table CLXXIX.—Properties of Saturated Steam—*continued.*

Pressure per Square inch from Mean Atmospheric Pressure. lbs.	Temperature in Fahrenheit Degrees.	Specific or Relative Volume of the Steam.	Density or Weight of 1 Cubic Foot of the Steam. lb.	Cubic Feet of the Steam per lb.	Latent Heat of Evaporation in Thermal Units per lb. of the Steam.	Total Heat in Thermal Units from 32° Fahrenheit per lb. of the Steam.	Absolute Pressure per Square Inch. lbs.
205	389·4	129·3	·4824	2·073	838·9	1200·7	219·7
210	391·3	126·5	·4926	2·030	837·5	1201·3	224·7
215	393·2	123·9	·5027	1·989	836·2	1201·9	229·7
220	395·1	121·5	·5128	1·950	834·8	1202·5	234·7
225	397·0	119·2	·5230	1·912	833·4	1203·1	239·7
230	398·9	117·0	·5332	1·876	832·1	1203·7	244·7
235	400·6	114·8	·5435	1·840	830·8	1204·2	249·7
240	402·3	112·6	·5538	1·805	829·5	1204·7	254·7
245	404·0	110·5	·5641	1·772	828·2	1205·2	259·7
250	405·7	108·6	·5744	1·741	826·9	1205·8	264·7
255	407·4	106·7	·5846	1·711	825·7	1206·3	269·7
260	409·0	104·8	·5949	1·681	824·5	1206·8	274·7
265	410·7	103·0	·6052	1·652	823·3	1207·3	279·7
270	412·3	101·3	·6155	1·624	822·1	1207·8	284·7
275	413·9	99·7	·6258	1·597	820·9	1208·3	289·7
280	415·5	98·2	·6361	1·572	819·7	1208·8	294·7
285	417·0	96·7	·6462	1·548	818·6	1209·2	299·7
290	418·5	95·2	·6563	1·524	817·5	1209·7	304·7
295	420·0	93·7	·6664	1·501	816·4	1210·2	309·7
300	421·5	92·3	·6766	1·478	815·3	1210·6	314·7
305	423·0	90·9	·6868	1·456	814·2	1211·0	319·7
310	424·4	89·6	·6968	1·435	813·1	1211·5	324·7
315	425·8	88·3	·7068	1·415	812·1	1211·9	329·7
320	427·2	87·0	·7168	1·395	811·1	1212·3	334·7
325	428·6	85·8	·7268	1·376	810·1	1212·7	339·7
350	435·4	80·3	·7770	1·287	805·0	1214·8	364·7
400	447·9	71·2	·8764	1·141	795·7	1218·6	414·7
450	459·3	63·9	·9756	1·025	787·2	1222·1	464·7
500	469·9	58·1	1·0741	·931	779·2	1225·3	514·7

Table CLXXIXa.—Total Heat of Evaporation from and at Differing Temperatures.

From	At 328° Press. 100	At 335° Press. 110	At 341° Press. 120	At 347° Press. 130.	At 353° Press. 140	At 358° Press. 150.	At 363° Press. 160	At 368° Press. 170	At 373° Press. 180	At 377° Press. 190	At 381° Press. 200	At 396° Press. 210	At 389° Press. 220	At 393° Press. 230
40° F.	1174	1176	1178	1180	1182	1183	1185	1186	1188	1189	1191	1192	1193	1194
50°	1164	1166	1168	1170	1172	1173	1175	1176	1178	1179	1181	1182	1183	1184
60°	1154	1156	1158	1160	1162	1163	1165	1166	1168	1169	1171	1172	1173	1174
70°	1144	1146	1148	1150	1152	1153	1155	1156	1158	1159	1161	1162	1163	1164
80°	1134	1136	1138	1140	1142	1143	1145	1146	1148	1149	1151	1152	1153	1154
90°	1124	1126	1128	1130	1132	1133	1135	1136	1138	1139	1141	1142	1143	1144
100°	1114	1116	1118	1120	1122	1123	1125	1126	1128	1129	1131	1132	1133	1134
110°	1104	1106	1108	1110	1112	1113	1115	1116	1118	1119	1121	1122	1123	1124
120°	1094	1096	1098	1100	1102	1103	1105	1106	1108	1109	1111	1112	1113	1114
130°	1084	1086	1088	1090	1092	1093	1095	1096	1098	1099	1101	1102	1103	1104
140°	1074	1076	1078	1080	1082	1083	1085	1086	1088	1089	1091	1092	1093	1094
150°	1064	1066	1068	1070	1072	1073	1075	1076	1078	1079	1081	1082	1083	1084
160°	1054	1056	1058	1060	1062	1063	1065	1066	1068	1069	1071	1072	1073	1074
170°	1044	1046	1048	1050	1052	1053	1055	1056	1058	1059	1061	1062	1063	1064
180°	1034	1036	1038	1040	1042	1043	1045	1046	1048	1049	1051	1052	1053	1054
190°	1024	1026	1028	1030	1032	1033	1035	1036	1038	1039	1041	1042	1043	1044
200°	1014	1016	1018	1020	1022	1023	1025	1026	1028	1029	1031	1032	1033	1034
210°	1004	1006	1008	1010	1012	1013	1015	1016	1018	1019	1021	1022	1023	1024
212°	1002	1004	1006	1008	1010	1011	1013	1014	1016	1017	1019	1020	1021	1022
220°	994	996	998	1000	1002	1003	1005	1006	1008	1009	1011	1012	1013	1014
230°	984	986	988	990	992	993	995	996	998	999	1001	1002	1003	1004
240°	974	976	978	980	982	983	985	986	988	989	991	992	993	994
250°	964	966	968	970	972	973	975	976	978	979	981	982	983	984

N.B.—Pressures are in lbs. absolute; temperatures degrees Fahrenheit.

ENGLISH AND METRICAL MEASURES.

The English standard yard is on a bronze bar, measured at a temperature of 62° F.

The French standard metre is the length of a platinum bar when at a temperature of 0° Cent. (32° F.).

The Standards Commission, in their Report of 1871–72, considered that a correction was needed to allow for this difference, but the original equivalents were adopted without correction in the Weights and Measures Act of 1878.

Comparison of	Metre.	Litre.	Kilogramme.
	Inches.*	Gallons.†	Lbs.
Corrected (Standards Com.), .	39·38202	·22018	2·20462
Adopted (Act of 1878), . .	39·37079	·2200967	2·20462

Square Measure.

ins.	feet.	yards.	perches.	roods.	acre.	square metres.
1 =	·00694	= ·000772 =	·0000255 =	·00000064 =	·000000159 =	·000645
144 =	1	= ·111 =	·00367 =	·0000918 =	·000023 =	·0929
1296 =	9	= 1 =	·0331 =	·000826 =	·0002062 =	·8361
39204 =	272¼	= 30¼ =	1 =	·025 =	·00625 =	25·292
1568160 =	10890	= 1210 =	40 =	1 =	·25 =	1011·7
6272640 =	43560	= 4840 =	160 =	4 =	1 =	4046·7

10 square chains = 1 acre.

1 hectare = 2·471143 acres.

1 square mile = 27878400 sq. feet. = 3097600 sq. yards. = 640 acres.

Cubic Measure.

ins.	feet.	yard.	cubic metre or stere.
1 =	·0005788 =	·000002144 =	·00016386
1728 =	1 =	·03704 =	·028315
46656 =	27 =	1 =	·764513

Measure of Capacity.

pints.	gall	peck.	bushel.	quarter.	wey.	last.	cub. ft.	litres.
1 =	·125 =	·0625 =	·01562 =	·00195 =	·00039 =	·000195 =	·020051 =	·5679
8 =	1 =	·5 =	·125 =	·0156 =	·00312 =	·00156 =	·16046 =	4·543
16 =	2 =	1 =	·25 =	·03125 =	·0625 =	·00312 =	·32092 =	9·087
64 =	8 =	4 =	1 =	·125 =	·025 =	·0125 =	1·28367 =	36·34766
512 =	64 =	32 =	8 =	1 =	·2 =	·1 =	10·269 =	290·781
2560 =	320 =	160 =	40 =	5 =	1 =	·5 =	51·347 =	1453·906
5120 =	640 =	320 =	80 =	10 =	2 =	1 =	102·69 =	2907·81

1 gallon in wine, ale, or dry measure = 277·27384 cubic inches = ·16 cubic foot. = 10 lbs. of distilled water =

Cube feet × 6·2355 = gallons × 10 = lbs.

Cube ins. × ·003607 = gallons.

1 bushel = 2218·19 cubic inches = 1·28 cubic foot.

Cubic feet × ·78 = bushels.

* At equal temperatures in ordinary air.

† At equal temperatures,—distilled water.

Table CLXXX.—Nautical Miles (Knots),* Miles, Kilometres, &c.

Speeds per hour.			Feet per minute.	Feet per second.	Speeds per hour.			Feet per minute.	Feet per second.
Admiralty Knots.	Statute Miles.	Kilometres.			Admiralty Knots.	Statute Miles.	Kilometres.		
4	4·6061	7·413	405·3	6·75	10	11·5151	18·532	1013·3	16·89
4·25	4·8939	7·876	430·7	7·17	10·25	11·8030	18·995	1038·7	17·31
4·5	5·1818	8·339	456	7·59	10·5	12·0909	19·458	1064	17·73
4·75	5·4697	8·802	481·3	8·02	10·75	12·3788	19·921	1089·3	18·15
5	5·7576	9·266	506·7	8·44	11	12·6667	20·385	1114·7	18·58
5·25	6·0454	9·729	532	8·86	11·25	12·9545	20·848	1140	19·00
5·5	6·3333	10·192	557·3	9·28	11·5	13·2424	21·311	1165·3	19·42
5·75	6·6212	10·656	582·7	9·71	11·75	13·5303	21·775	1190·7	19·85
6	6·9091	11·119	608	10·13	12	13·8182	22·238	1216	20·27
6·25	7·1970	11·582	633·3	10·55	12·25	14·1061	22·701	1241·3	20·69
6·5	7·4848	12·046	658·7	10·97	12·5	14·3939	23·165	1266·7	21·11
6·75	7·7727	12·509	684	11·40	12·75	14·6818	23·628	1292	21·54
7	8·0606	12·927	709·3	11·82	13	14·9697	24·091	1317·3	21·95
7·25	8·3485	13·435	734·7	12·24	13·25	15·2576	24·554	1342·7	22·38
7·5	8·6364	13·899	760	12·66	13·5	15·5454	25·018	1368	22·80
7·75	8·9242	14·362	785·3	13·09	13·75	15·8333	25·481	1393·3	23·22
8	9·2121	14·825	810·7	13·51	14	16·1212	25·944	1418·7	23·64
8·25	9·5000	15·289	836	13·93	14·25	16·4091	26·408	1444	24·07
8·5	9·7879	15·752	861·3	14·35	14·5	16·6970	26·871	1469·3	24·49
8·75	10·0757	16·215	886·7	14·78	14·75	16·9848	27·334	1494·7	24·91
9	10·3636	16·678	912	15·20	15	17·2727	27·797	1520	25·33
9·25	10·6515	17·142	937·3	15·62	15·25	17·5606	28·261	1545·3	25·76
9·5	10·9394	17·605	962·7	16·04	15·5	17·8485	28·724	1570·7	26·18
9·75	11·2273	18·068	988	16·47	15·75	18·1364	29·187	1598	26·60

* *N.B.*—Strictly speaking, a knot is a rate of speed and not a measure of distance.

Table CLXXX.—Nautical Miles (Knots), Miles, Kilometres, &c.—*continued.*

16	18·4242	29·651	1621·3	27·02	23·25	26·7727	43·086	2356	39·27
16·25	18·7121	30·114	1646·7	27·45	23·5	27·0606	43·549	2381·3	39·69
16·5	19·0000	30·577	1672	27·87	23·75	27·3485	44·013	2406·7	40·11
16·75	19·2879	31·041	1697·3	28·29	24	27·6364	44·476	2432	40·53
17	19·5757	31·504	1722·7	28·71	24·25	27·9242	44·939	2457·3	40·95
17·25	19·8636	31·967	1748	29·13	24·5	28·2121	45·403	2482·7	41·38
17·5	20·1515	32·430	1773·3	29·55	24·75	28·5000	45·866	2508	41·80
17·75	20·4394	32·894	1798·7	29·98	25	28·7879	46·329	2533·3	42·22
18	20·7273	33·357	1824	30·40	25·25	29·0757	46·792	2558·7	42·64
18·25	21·0151	33·820	1849·3	30·82	25·5	29·3636	47·256	2584	43·07
18·5	21·3030	34·284	1874·7	31·24	25·75	29·6515	47·719	2609·3	43·49
18·75	21·5909	34·747	1900	31·67	26	29·9394	48·182	2634·7	43·91
19	21·8788	35·210	1925·3	32·09	26·25	30·2273	48·646	2660	44·33
19·25	22·1667	35·673	1950·7	32·51	26·5	30·5151	49·109	2685·3	44·75
19·5	22·4545	36·137	1976	32·93	26·75	30·8030	49·572	2710·7	45·18
19·75	22·7424	36·600	2001·3	33·35	27	31·0909	50·035	2736	45·60
20	23·0303	37·063	2026·7	33·78	27·25	31·3788	50·499	2761·3	46·02
20·25	23·3182	37·526	2052	34·20	27·5	31·6667	50·962	2786·7	46·44
20·5	23·6061	37·990	2077·3	34·62	27·75	31·9545	51·425	2812	46·87
20·75	23·8939	38·453	2102·7	35·04	28	32·2424	51·889	2837·3	47·29
21	24·1818	38·916	2128	35·47	28·25	32·5303	52·352	2862·7	47·71
21·25	24·4697	39·379	2153·3	35·89	28·5	32·8182	52·815	2888	48·13
21·5	24·7576	39·843	2178·7	36·31	28·75	33·1061	53·279	2913·3	48·55
21·75	25·0454	40·306	2204	36·73	29	33·3939	53·742	2938·7	48·98
22	25·3333	40·769	2229·3	37·15	29·25	33·6818	54·205	2964	49·40
22·25	25·6212	41·233	2254·7	37·58	29·5	33·9697	54·668	2989·3	49·82
22·5	25·9091	41·696	2280	38·00	29·75	34·2576	55·132	3014·7	50·24
22·75	26·1970	42·160	2305·3	38·42	30	34·5454	55·595	3040	50·67
23	26·4848	42·623	2330·7	38·84	30·25	34·8333	56·058	3065·3	51·09

Table CLXXX.—Nautical Miles (Knots), Miles, Kilometres, &c.—*continued.*

Speeds per hour.			Feet per minute.	Feet per second.	Speeds per hour.			Feet per minute.	Feet per second.
Admiralty Knots.	Statute Miles.	Kilometres.			Admiralty Knots.	Statute Miles.	Kilometres.		
30·5	35·1212	56·522	3090·7	51·52	34·25	39·4393	63·471	3470·7	57·84
30·75	35·4091	56·985	3116	51·94	34·5	39·7272	63·934	3496	58·26
31	35·6970	57·448	3141·3	52·36	34·75	40·0151	64·397	3521·3	58·68
31·25	35·9849	57·911	3166·7	52·78	35	40·3030	64·860	3546·7	59·10
31·5	36·2728	58·374	3192	53·20	35·25	40·5909	65·324	3572	59·53
31·75	36·5606	58·838	3217·3	53·62	35·5	40·8788	65·788	3597·3	59·96
32	36·8484	59·302	3242·7	54·04	35·75	41·1667	66·251	3622·7	60·38
32·25	37·1363	59·765	3268	54·47	36	41·4546	66·714	3648	60·80
32·5	37·4242	60·228	3293·3	54·90	36·25	41·7424	67·177	3673·3	61·22
32·75	37·7121	60·691	3318·7	55·32	36·5	42·0302	67·640	3698·7	61·64
33	38·0000	61·154	3344	55·74	36·75	42·3181	68·104	3724	62·06
33·25	38·2879	61·618	3369·3	56·16	37	42·6060	68·568	3749·3	62·48
33·5	38·5758	62·082	3394·7	56·58	37·25	42·8939	69·031	3774·7	62·91
33·75	38·8636	62·545	3420	57·00	37·5	43·1818	69·494	3800	63·34
34	39·1514	63·008	3445·3	57·42	37·75	43·4697	69·957	3825·3	63·77

	Statute Mile.	Kilometre.	Feet per min.	Foot per sec.
$\frac{1}{10}$ Knot =	·11515	·1853	10·133	·1688
$\frac{1}{100}$ „ =	·011515	·01853	1·0133	·01688

Table CLXXX.—Nautical Miles (Knots), Miles, Kilometres, &c.—*continued.*

16	18·4242	29·651	1621·3	27·02	23·25	26·7727	43·086	2356	39·27
16·25	18·7121	30·114	1646·7	27·45	23·5	27·0606	43·549	2381·3	39·69
16·5	19·0000	30·577	1672	27·87	23·75	27·3485	44·013	2406·7	40·11
16·75	19·2879	31·041	1697·3	28·29	24	27·6364	44·476	2432	40·53
17	19·5757	31·504	1722·7	28·71	24·25	27·9242	44·939	2457·3	40·95
17·25	19·8636	31·967	1748	29·13	24·5	28·2121	45·403	2482·7	41·38
17·5	20·1515	32·430	1773·3	29·55	24·75	28·5000	45·866	2508	41·80
17·75	20·4394	32·894	1798·7	29·98	25	28·7879	46·329	2533·3	42·22
18	20·7273	33·357	1824	30·40	25·25	29·0757	46·792	2558·7	42·64
18·25	21·0151	33·820	1849·3	30·82	25·5	29·3636	47·256	2584	43·07
18·5	21·3030	34·284	1874·7	31·24	25·75	29·6515	47·719	2609·3	43·49
18·75	21·5909	34·747	1900	31·67	26	29·9394	48·182	2634·7	43·91
19	21·8788	35·210	1925·3	32·09	26·25	30·2273	48·646	2660	44·33
19·25	22·1667	35·673	1950·7	32·51	26·5	30·5151	49·109	2685·3	44·75
19·5	22·4545	36·137	1976	32·93	26·75	30·8030	49·572	2710·7	45·18
19·75	22·7424	36·600	2001·3	33·35	27	31·0909	50·035	2736	45·60
20	23·0303	37·063	2026·7	33·78	27·25	31·3788	50·499	2761·3	46·02
20·25	23·3182	37·526	2052	34·20	27·5	31·6667	50·962	2786·7	46·44
20·5	23·6061	37·990	2077·3	34·62	27·75	31·9545	51·425	2812	46·87
20·75	23·8939	38·453	2102·7	35·04	28	32·2424	51·889	2837·3	47·29
21	24·1818	38·916	2128	35·47	28·25	32·5303	52·352	2862·7	47·71
21·25	24·4697	39·379	2153·3	35·89	28·5	32·8182	52·815	2888	48·13
21·5	24·7576	39·843	2178·7	36·31	28·75	33·1061	53·279	2913·3	48·55
21·75	25·0454	40·306	2204	36·73	29	33·3939	53·742	2938·7	48·98
22	25·3333	40·769	2229·3	37·15	29·25	33·6818	54·205	2964	49·40
22·25	25·6212	41·233	2254·7	37·58	29·5	33·9697	54·668	2989·3	49·82
22·5	25·9091	41·696	2280	38·00	29·75	34·2576	55·132	3014·7	50·24
22·75	26·1970	42·160	2305·3	38·42	30	34·5454	55·595	3040	50·67
23	26·4848	42·623	2330·7	38·84	30·25	34·8333	56·058	3065·3	51·09

Table CLXXX.—Nautical Miles (Knots), Miles, Kilometres, &c.—*continued.*

Speeds per hour.			Feet per minute.	Feet per second.	Speeds per hour.			Feet per minute.	Feet per second.
Admiralty Knots.	Statute Miles.	Kilometres.			Admiralty Knots.	Statute Miles.	Kilometres.		
30·5	35·1212	56·522	3090·7	51·52	34·25	39·4393	63·471	3470·7	57·84
30·75	35·4091	56·985	3116	51·94	34·5	39·7272	63·934	3496	58·26
31	35·6970	57·448	3141·3	52·36	34·75	40·0151	64·397	3521·3	58·68
31·25	35·9849	57·911	3166·7	52·78	35	40·3030	64·860	3546·7	59·10
31·5	36·2728	58·374	3192	53·20	35·25	40·5909	65·324	3572	59·53
31·75	36·5606	58·838	3217·3	53·62	35·5	40·8788	65·788	3597·3	59·96
32	36·8484	59·302	3242·7	54·04	35·75	41·1667	66·251	3622·7	60·38
32·25	37·1363	59·765	3268	54·47	36	41·4546	66·714	3648	60·80
32·5	37·4242	60·228	3293·3	54·90	36·25	41·7424	67·177	3673·3	61·22
32·75	37·7121	60·691	3318·7	55·32	36·5	42·0302	67·640	3698·7	61·64
33	38·0000	61·154	3344	55·74	36·75	42·3181	68·104	3724	62·06
33·25	38·2879	61·618	3369·3	56·16	37	42·6060	68·568	3749·3	62·48
33·5	38·5758	62·082	3394·7	56·58	37·25	42·8939	69·031	3774·7	62·91
33·75	38·8636	62·545	3420	57·00	37·5	43·1818	69·494	3800	63·34
34	39·1514	63·008	3445·3	57·42	37·75	43·4697	69·957	3825·3	63·77

	Statute Mile.	Kilometre.	Feet per min.	Foot per sec.
$\frac{1}{10}$ Knot =	·11515	·1853	10·133	·1688
$\frac{1}{100}$,, =	·011515	·01853	1·0133	·01688

Table CLXXXI.—Kilometres and Admiralty Knots.

Kilometres.	Admiralty Knots.	Kilometres.	Admiralty Knots.	Kilometres.	Admiralty Knots.	Kilometres.	Admiralty Knots.
7·5	4·047	23·5	12·681	39·5	21·315	55·5	29·949
8	4·317	24	12·951	40	21·585	56	30·218
8·5	4·587	24·5	13·221	40·5	21·854	56·5	30·488
9	4·857	25	13·490	41	22·124	57	30·758
9·5	5·126	25·5	13·760	41·5	22·394	57·5	31·028
10	5·396	26	14·030	42	22·664	58	31·298
10·5	5·666	26·5	14·300	42·5	22·934	58·5	31·568
11	5·936	27	14·570	43	23·203	59	31·838
11·5	6·206	27·5	14·839	43·5	23·473	59·5	32·107
12	6·475	28	15·109	44	23·743	60	32·376
12·5	6·745	28·5	15·379	44·5	24·013	60·5	32·646
13	7·015	29	15·649	45	24·283	61	32·916
13·5	7·285	29·5	15·919	45·5	24·552	61·5	33·186
14	7·555	30	16·188	46	24·822	62	33·456
14·5	7·824	30·5	16·458	46·5	25·092	62·5	33·726
15	8·094	31	16·728	47	25·362	63	33·996
15·5	8·364	31·5	16·998	47·5	25·632	63·5	34·266
16	8·634	32	17·268	48	25·901	64	34·536
16·5	8·904	32·5	17·538	48·5	26·171	64·5	34·806
17	9·173	33	17·807	49	26·441	65	35·076
17·5	9·443	33·5	18·077	49·5	26·711	65·5	35·345
18	9·713	34	18·347	50	26·981	66	35·614
18·5	9·983	34·5	18·617	50·5	27·251	66·5	35·884
19	10·253	35	18·887	51	27·520	67	36·154
19·5	10·523	35·5	19·156	51·5	27·790	67·5	36·424
20	10·792	36	19·426	52	28·060	68	36·694
20·5	11·062	36·5	19·696	52·5	28·330	68·5	36·964
21	11·332	37	19·966	53	28·600	69	37·234
21·5	11·602	37·5	20·235	53·5	28·869	69·5	37·504
22	11·872	38	20·505	54	29·139	70	37·774
22·5	12·141	38·5	20·775	54·5	29·409	70·5	38·044
23	12·411	39	21·045	55	29·679	71·	38·314

$\frac{1}{10}$ Kilometre per hour = ·05396 of an Admiralty knot.
$\frac{1}{100}$ " " " = ·005396 " " "
1 Admiralty knot " = 6080 ft. per hour.
1 Statute mile = 5280 ft.
1 Kilometre = 3280·8992 ft.

Table CLXXXII.—Millimetres and Inches.

Millimetres.	Inches.	Millimetres.	Inches.	Millimetres.	Inches.	Millimetres.	Inches.	Millimetres.	Inches.
1	·03937	41	1·6142	81	3·1890	121	4·7639	161	6·3387
2	·07874	42	1·6536	82	3·2284	122	4·8032	162	6·3781
3	·11811	43	1·6929	83	3·2678	123	4·8426	163	6·4174
4	·15748	44	1·7323	84	3·3071	124	4·8820	164	6·4568
5	·19685	45	1·7717	85	3·3465	125	4·9214	165	6·4962
6	·23622	46	1·8110	86	3·3859	126	4·9607	166	6·5356
7	·27560	47	1·8504	87	3·4252	127	5·0001	167	6·5749
8	·31497	48	1·8898	88	3·4646	128	5·0395	168	6·6143
9	·35434	49	1·9292	89	3·5040	129	5·0788	169	6·6537
10	·3937	50	1·9685	90	3·5434	130	5·1182	170	6·6930
11	·4331	51	2·0079	91	3·5827	131	5·1576	171	6·7324
12	·4724	52	2·0473	92	3·6221	132	5·1969	172	6·7718
13	·5118	53	2·0866	93	3·6614	133	5·2363	173	6·8111
14	·5512	54	2·1260	94	3·7008	134	5·2757	174	6·8505
15	·5906	55	2·1654	95	3·7402	135	5·3150	175	6·8899
16	·6299	56	2·2048	96	3·7796	136	5·3544	176	6·9293
17	·6693	57	2·2441	97	3·8190	137	5·3938	177	6·9686
18	·7087	58	2·2835	98	3·8583	138	5·4332	178	7·0080
19	·7480	59	2·3229	99	3·8977	139	5·4725	179	7·0474
20	·7874	60	2·3622	100	3·9371	140	5·5119	180	7·0867
21	·8268	61	2·4016	101	3·9764	141	5·5513	181	7·1261
22	·8662	62	2·4410	102	4·0158	142	5·5906	182	7·1655
23	·9055	63	2·4804	103	4·0552	143	5·6300	183	7·2048
24	·9449	64	2·5197	104	4·0946	144	5·6694	184	7·2442
25	·9843	65	2·5591	105	4·1339	145	5·7088	185	7·2836
26	1·0236	66	2·5985	106	4·1733	146	5·7481	186	7·3230
27	1·0630	67	2·6378	107	4·2127	147	5·7875	187	7·3623
28	1·1024	68	2·6772	108	4·2520	148	5·8269	188	7·4017
29	1·1418	69	2·7166	109	4·2914	149	5·8662	189	7·4411
30	1·1811	70	2·7560	110	4·3308	150	5·9056	190	7·4804
31	1·2205	71	2·7953	111	4·3702	151	5·9450	191	7·5198
32	1·2599	72	2·8347	112	4·4095	152	5·9844	192	7·5592
33	1·2992	73	2·8741	113	4·4489	153	6·0237	193	7·5986
34	1·3386	74	2·9134	114	4·4883	154	6·0631	194	7·6379
35	1·3780	75	2·9528	115	4·5276	155	6·1025	195	7·6773
36	1·4173	76	2·9922	116	4·5670	156	6·1418	196	7·7167
37	1·4567	77	3·0316	117	4·6064	157	6·1812	197	7·7560
38	1·4961	78	3·0709	118	4·6458	158	6·2206	198	7·7954
39	1·5355	79	3·1103	119	4·6851	159	6·2600	199	7·8348
40	1·5748	80	3·1497	120	4·7245	160	6·2993	200	7·8742

Table CLXXXII.—Millimetres and Inches—*continued.*

Millimetres.	Inches.	Millimetres.	Inches.	Millimetres.	Inches.	Millimetres.	Inches.	Millimetres.	Inches.
201	7·9135	241	9·4884	281	11·0632	321	12·6380	361	14·2128
202	7·9529	242	9·5277	282	11·1026	322	12·6774	362	14·2522
203	7·9923	243	9·5671	283	11·1419	323	12·7168	363	14·2916
204	8·0316	244	9·6065	284	11·1813	324	12·7561	364	14·3310
205	8·0710	245	9·6458	285	11·2207	325	12·7955	365	14·3703
206	8·1104	246	9·6852	286	11·2600	326	12·8349	366	14·4097
207	8·1498	247	9·7246	287	11·2994	327	12·8742	367	14·4491
208	8·1891	248	9·7640	288	11·3388	328	12·9136	368	14·4884
209	8·2285	249	9·8033	289	11·3782	329	12·9530	369	14·5278
210	8·2679	250	9·8427	290	11·4175	330	12·9924	370	14·5672
211	8·3072	251	9·8821	291	11·4569	331	13·0317	371	14·6066
212	8·3466	252	9·9214	292	11·4963	332	13·0711	372	14·6459
213	8·3860	253	9·9608	293	11·5356	333	13·1105	373	14·6853
214	8·4254	254	10·0002	294	11·5750	334	13·1498	374	14·7247
215	8·4647	255	10·0396	295	11·6144	335	13·1892	375	14·7640
216	8·5041	256	10·0789	296	11·6538	336	13·2286	376	14·8034
217	8·5435	257	10·1183	297	11·6931	337	13·2680	377	14·8428
218	8·5828	258	10·1577	298	11·7325	338	13·3073	378	14·8822
219	8·6222	259	10·1970	299	11·7719	339	13·3467	379	14·9215
220	8·6616	260	10·2364	300	11·8112	340	13·3861	380	14·9609
221	8·7009	261	10·2758	301	11·8506	341	13·4254	381	15·0003
222	8·7403	262	10·3151	302	11·8900	342	13·4648	382	15·0396
223	8·7797	263	10·3545	303	11·9294	343	13·5042	383	15·0790
224	8·8190	264	10·3939	304	11·9687	344	13·5436	384	15·1184
225	8·8584	265	10·4333	305	12·0081	345	13·5829	385	15·1578
226	8·8978	266	10·4726	306	12·0475	346	13·6223	386	15·1971
227	8·9372	267	10·5120	307	12·0868	347	13·6617	387	15·2365
228	8·9765	268	10·5514	308	12·1262	348	13·7010	388	15·2759
229	9·0159	269	10·5907	309	12·1656	349	13·7404	389	15·3152
230	9·0553	270	10·6301	310	12·2049	350	13·7798	390	15·3546
231	9·0946	271	10·6695	311	12·2443	351	13·8192	391	15·3940
232	9·1340	272	10·7088	312	12·2837	352	13·8585	392	15·4334
233	9·1734	273	10·7482	313	12·3231	353	13·8979	393	15·4727
234	9·2128	274	10·7876	314	12·3624	354	13·9373	394	15·5121
235	9·2521	275	10·8270	315	12·4018	355	13·9766	395	15·5515
236	9·2915	276	10·8663	316	12·4412	356	14·0160	396	15·5908
237	9·3309	277	10·9057	317	12·4805	357	14·0554	397	15·6302
238	9·3702	278	10·9451	318	12·5199	358	14·0947	398	15·6696
239	9·4096	279	10·9844	319	12·5593	359	14·1341	399	15·7089
240	9·4490	280	11·0238	320	12·5986	360	14·1735	400	15·7483

Table CLXXXII.—Millimetres and Inches—*continued.*

Milli-metres.	Inches.	Milli-metres.	Inches.	Milli-metres.	Inches.	Milli-metres.	Inches.	Milli-metres.	Inches.
401	15·7877	441	17·3625	481	18·9374	521	20·5122	561	22·0870
402	15·8271	442	17·4019	482	18·9767	522	20·5516	562	22·1264
403	15·8664	443	17·4413	483	19·0161	523	20·5909	563	22·1658
404	15·9058	444	17·4806	484	19·0555	524	20·6303	564	22·2051
405	15·9452	445	17·5200	485	19·0948	525	20·6697	565	22·2445
406	15·9845	446	17·5594	486	19·1342	526	20·7090	566	22·2839
407	16·0239	447	17·5987	487	19·1736	527	20·7484	567	22·3232
408	16·0633	448	17·6381	488	19·2130	528	20·7878	568	22·3626
409	16·1026	449	17·6775	489	19·2523	529	20·8272	569	22·4020
410	16·1420	450	17·7169	490	19·2917	530	20·8665	570	22·4414
411	16·1814	451	17·7562	491	19·3311	531	20·9059	571	22·4807
412	16·2208	452	17·7956	492	19·3704	532	20·9453	572	22·5201
413	16·2601	453	17·8350	493	19·4098	533	20·9846	573	22·5595
414	16·2995	454	17·8743	494	19·4492	534	21·0240	574	22·5988
415	16·3389	455	17·9137	495	19·4885	535	21·0634	575	22·6382
416	16·3782	456	17·9531	496	19·5279	536	21·1027	576	22·6776
417	16·4176	457	17·9924	497	19·5673	537	21·1421	577	22·7170
418	16·4570	458	18·0318	498	19·6066	538	21·1815	578	22·7563
419	16·4964	459	18·0712	499	19·6460	539	21·2209	579	22·7957
420	16·5357	460	18·1106	500	19·6854	540	21·2602	580	22·8351
421	16·5751	461	18·1499	501	19·7248	541	21·2996	581	22·8744
422	16·6145	462	18·1893	502	19·7641	542	21·3390	582	22·9138
423	16·6538	463	18·2287	503	19·8035	543	21·3783	583	22·9532
424	16·6932	464	18·2680	504	19·8429	544	21·4177	584	22·9925
425	16·7326	465	18·3074	505	19·8822	545	21·4571	585	23·0319
426	16·7720	466	18·3468	506	19·9216	546	21·4964	586	23·0713
427	16·8113	467	18·3862	507	19·9610	547	21·5358	587	23·1106
428	16·8507	468	18·4255	508	20·0004	548	21·5752	588	23·1500
429	16·8901	469	18·4649	509	20·0397	549	21·6146	589	23·1894
430	16·9294	470	18·5043	510	20·0791	550	21·6539	590	23·2288
431	16·9688	471	18·5436	511	20·1185	551	21·6933	591	23·2681
432	17·0082	472	18·5830	512	20·1578	552	21·7327	592	23·3075
433	17·0476	473	18·6224	513	20·1972	553	21·7720	593	23·3469
434	17·0869	474	18·6617	514	20·2366	554	21·8114	594	23·3862
435	17·1263	475	18·7011	515	20·2760	555	21·8508	595	23·4256
436	17·1657	476	18·7405	516	20·3153	556	21·8902	596	23·4650
437	17·2050	477	18·7799	517	20·3547	557	21·9295	597	23·5044
438	17·2444	478	18·8192	518	20·3941	558	21·9689	598	23·5437
439	17·2838	479	18·8586	519	20·4334	559	22·0083	599	23·5831
440	17·3232	480	18·8980	520	20·4728	560	22·0476	600	23·6225

Table CLXXXII.—Millimetres and Inches—*continued.*

Millimetres.	Inches.	Millimetres.	Inches.	Millimetres.	Inches.	Millimetres.	Inches.	Millimetres.	Inches.
601	23·6618	641	25·2367	681	26·8115	721	28·3863	761	29·9612
602	23·7012	642	25·2760	682	26·8509	722	28·4257	762	30·0005
603	23·7406	643	25·3154	683	26·8902	723	28·4651	763	30·0399
604	23·7800	644	25·3548	684	26·9296	724	28·5044	764	30·0793
605	23·8193	645	25·3942	685	26·9690	725	28·5438	765	30·1187
606	23·8587	646	25·4335	686	27·0084	726	28·5832	766	30·1580
607	23·8981	647	25·4729	687	27·0477	727	28·6226	767	30·1974
608	23·9374	648	25·5123	688	27·0871	728	28·6619	768	30·2368
609	23·9768	649	25·5516	689	27·1265	729	28·7013	769	30·2761
610	24·0162	650	25·5910	690	27·1658	730	28·7407	770	30·3155
611	24·0556	651	25·6304	691	27·2052	731	28·7800	771	30·3549
612	24·0949	652	25·6698	692	27·2446	732	28·8194	772	30·3942
613	24·1343	653	25·7091	693	27·2840	733	28·8588	773	30·4336
614	24·1737	654	25·7485	694	27·3233	734	28·8982	774	30·4730
615	24·2130	655	25·7879	695	27·3627	735	28·9375	775	30·5124
616	24·2524	656	25·8272	696	27·4021	736	28·9769	776	30·5517
617	24·2918	657	25·8666	697	27·4414	737	29·0163	777	30·5911
618	24·3312	658	25·9060	698	27·4808	738	29·0556	778	30·6305
619	24·3705	659	25·9454	699	27·5202	739	29·0950	779	30·6698
620	24·4099	660	25·9847	700	27·5596	740	29·1344	780	30·7092
621	24·4493	661	26·0241	701	27·5989	741	29·1738	781	30·7486
622	24·4886	662	26·0635	702	27·6383	742	29·2131	782	30·7880
623	24·5280	663	26·1028	703	27·6777	743	29·2525	783	30·8273
624	24·5674	664	26·1422	704	27·7170	744	29·2919	784	30·8667
625	24·6068	665	26·1816	705	27·7564	745	29·3312	785	30·9061
626	24·6461	666	26·2210	706	27·7958	746	29·3706	786	30·9454
627	24·6855	667	26·2603	707	27·8352	747	29·4100	787	30·9848
628	24·7249	668	26·2997	708	27·8745	748	29·4494	788	31·0242
629	24·7642	669	26·3391	709	27·9139	749	29·4887	789	31·0636
630	24·8036	670	26·3784	710	27·9533	750	29·5281	790	31·1029
631	24·8430	671	26·4178	711	27·9926	751	29·5675	791	31·1423
632	24·8823	672	26·4572	712	28·0320	752	29·6068	792	31·1817
633	24·9217	673	26·4965	713	28·0714	753	29·6462	793	31·2210
634	24·9611	674	26·5359	714	28·1108	754	29·6856	794	31·2604
635	25·0004	675	26·5753	715	28·1501	755	29·7250	795	31·2998
636	25·0398	676	26·6147	716	28·1895	756	29·7643	796	31·3392
637	25·0792	677	26·6540	717	28·2289	757	29·8037	797	31·3785
638	25·1186	678	26·6934	718	28·2682	758	29·8431	798	31·4179
639	25·1579	679	26·7328	719	28·3076	759	29·8824	799	31·4573
640	25·1973	680	26·7721	720	28·3470	760	29·9218	800	31·4966

Table CLXXXII.—Millimetres and Inches—*continued.*

Millimetres.	Inches.	Millimetres.	Inches.	Millimetres.	Inches.	Millimetres.	Inches.	Millimetres.	Inches.
801	31·5360	841	33·1108	881	34·6857	921	36·2605	961	37·8253
802	31·5754	842	33·1502	882	34·7250	922	36·2999	962	37·8747
803	31·6148	843	33·1896	883	34·7644	923	36·3392	963	37·9141
804	31·6541	844	33·2290	884	34·8038	924	36·3786	964	37·9534
805	31·6935	845	33·2683	885	34·8432	925	36·4180	965	37·9928
806	31·7329	846	33·3077	886	34·8825	926	36·4574	966	38·0322
807	31·7722	847	33·3471	887	34·9219	927	36·4967	967	38·0716
808	31·8116	848	33·3864	888	34·9613	928	36·5361	968	38·1109
809	31·8510	849	33·4258	889	35·0006	929	36·5755	969	38·1503
810	31·8903	850	33·4652	890	35·0400	930	36·6148	970	38·1897
811	31·9297	851	33·5046	891	35·0794	931	36·6542	971	38·2290
812	31·9691	852	33·5439	892	35·1188	932	36·6936	972	38·2684
813	32·0085	853	33·5833	893	35·1581	933	36·7330	973	38·3078
814	32·0478	854	33·6227	894	35·1975	934	36·7723	974	38·3472
815	32·0872	855	33·6620	895	35·2369	935	36·8117	975	38·3865
816	32·1266	856	33·7014	896	35·2762	936	36·8511	976	38·4259
817	32·1659	857	33·7408	897	35·3156	937	36·8904	977	38·4653
818	32·2053	858	33·7801	898	35·3550	938	36·9298	978	38·5046
819	32·2447	859	33·8195	899	35·3943	939	36·9692	979	38·5440
820	32·2840	860	33·8589	900	35·4337	940	37·0086	980	38·5834
821	32·3234	861	33·8982	901	35·4731	941	37·0479	981	38·6228
822	32·3628	862	33·9376	902	35·5125	942	37·0873	982	38·6621
823	32·4022	863	33·9770	903	35·5518	943	37·1267	983	38·7015
824	32·4415	864	34·0164	904	35·5912	944	37·1660	984	38·7409
825	32·4809	865	34·0557	905	35·6306	945	37·2054	985	38·7802
826	32·5203	866	34·0951	906	35·6699	946	37·2448	986	38·8196
827	32·5596	867	34·1345	907	35·7093	947	37·2841	987	38·8590
828	32·5990	868	34·1738	908	35·7487	948	37·3235	988	38·8984
829	32·6384	869	34·2132	909	35·7880	949	37·3629	989	38·9377
830	32·6778	870	34·2526	910	35·8274	950	37·4023	990	38·9771
831	32·7171	871	34·2920	911	35·8668	951	37·4416	991	39·0165
832	32·7565	872	34·3313	912	35·9062	952	37·4810	992	39·0558
833	32·7959	873	34·3707	913	35·9455	953	37·5204	993	39·0952
834	32·8352	874	34·4101	914	35·9849	954	37·5597	994	39·1346
835	32·8746	875	34·4494	915	36·0243	955	37·5991	995	39·1739
836	32·9140	876	34·4888	916	36·0636	956	37·6385	996	39·2133
837	32·9534	877	34·5282	917	36·1030	957	37·6778	997	39·2527
838	32·9927	878	34·5676	918	36·1424	958	37·7172	998	39·2920
839	33·0321	879	34·6069	919	36·1818	959	37·7566	999	39·3314
840	33·0715	880	34·6463	920	36·2211	960	37·7960	1000	39·3708

965 mm. = 37·9928 ins.: 9·65 metres = 379·928 ins.: 96·5 metres = 3799·28 ins.: &c.

Table CLXXXIIa.—English Feet and French Metres.

Feet.	Metres.	Feet.	Metres.	Feet.	Metres.	Feet.	Metres.	Feet.	Metres.	Feet.	Metres.	Feet.	Metres.
1·0	0·305	13·0	3·965	25	7·625	49	14·945	73	22·265	97	29·585	121	36·905
1·5	0·458	13·5	4·118	26	7·930	50	15·250	74	22·570	98	29·890	122	37·210
2·0	0·610	14·0	4·270	27	8·235	51	15·555	75	22·875	99	30·195	123	37·515
2·5	0·762	14·5	4·423	28	8·540	52	15·860	76	23·180	100	30·500	124	37·820
3·0	0·915	15·0	4·575	29	8·845	53	16·165	77	23·485	101	30·805	125	38·125
3·5	1·068	15·5	4·728	30	9·150	54	16·470	78	23·790	102	31·110	126	38·430
4·0	1·220	16·0	4·880	31	9·455	55	16·775	79	24·095	103	31·415	127	38·735
4·5	1·373	16·5	5·033	32	9·760	56	17·080	80	24·400	104	31·720	128	39·040
5·0	1·525	17·0	5·185	33	10·065	57	17·385	81	24·705	105	32·025	129	39·345
5·5	1·678	17·5	5·338	34	10·370	58	17·690	82	25·010	106	32·330	130	39·650
6·0	1·830	18·0	5·490	35	10·675	59	17·995	83	25·315	107	32·635	131	39·955
6·5	1·983	18·5	5·643	36	10·980	60	18·300	84	25·620	108	32·940	132	40·260
7·0	2·135	19·0	5·795	37	11·285	61	18·605	85	25·925	109	33·245	133	40·565
7·5	2·288	19·5	5·948	38	11·590	62	18·910	86	26·230	110	33·550	134	40·870
8·0	2·440	20·0	6·100	39	11·895	63	19·215	87	26·535	111	33·855	135	41·175
8·5	2·593	20·5	6·253	40	12·200	64	19·520	88	26·840	112	34·160	136	41·480
9·0	2·745	21·0	6·405	41	12·505	65	19·825	89	27·145	113	34·465	137	41·785
9·5	2·898	21·5	6·558	42	12·810	66	20·130	90	27·450	114	34·770	138	42·090
10·0	3·050	22·0	6·710	43	13·115	67	20·435	91	27·755	115	35·075	139	42·395
10·5	3·203	22·5	6·863	44	13·420	68	20·740	92	28·060	116	35·380	140	42·700
11·0	3·355	23·0	7·015	45	13·725	69	21·045	93	28·365	117	35·685	141	43·005
11·5	3·508	23·5	7·168	46	14·030	70	21·350	94	28·670	118	35·990	142	43·310
12·0	3·660	24·0	7·320	47	14·335	71	21·655	95	28·975	119	36·295	143	43·615
12·5	3·813	24·5	7·473	48	14·640	72	21·960	96	29·280	120	36·600	144	43·920

Table CLXXXIIa.—English Feet and French Metres—*continued.*

Feet.	Metres.	Feet.	Metres.	Feet.	Metres.	Feet.	Metres.	Feet.	Metres.	Feet.	Metres.	Feet.	Metres.
145	44·225	169	51·545	193	58·865	217	66·185	241	73·505	265	80·825	289	88·145
146	44·530	170	51·850	194	59·170	218	66·490	242	73·810	266	81·130	290	88·450
147	44·835	171	52·155	195	59·475	219	66·795	243	74·115	267	81·435	291	88·755
148	45·140	172	52·460	196	59·780	220	67·100	244	74·420	268	81·740	292	89·060
149	45·445	173	52·765	197	60·085	221	67·405	245	74·725	269	82·045	293	89·365
150	45·750	174	53·070	198	60·390	222	67·710	246	75·030	270	82·350	294	89·670
151	46·055	175	53·375	199	60·695	223	68·015	247	75·335	271	82·655	295	89·975
152	46·360	176	53·680	200	61·000	224	68·320	248	75·640	272	82·960	296	90·280
153	46·665	177	53·985	201	61·305	225	68·625	249	75·945	273	83·265	297	90 585
154	46·970	178	54·290	202	61·610	226	68·930	250	76·250	274	83·570	298	90·890
155	47·275	179	54·595	203	61·915	227	69·235	251	76·555	275	83·875	299	91·195
156	47·580	180	54·900	204	62·220	228	69·540	252	76·860	276	84·180	300	91·500
157	47·885	181	55·205	205	62·525	229	69·845	253	77·165	277	84·485	301	91·805
158	48·190	182	55·510	206	62·830	230	70·150	254	77·470	278	84·790	302	92·110
159	48·495	183	55·815	207	63·135	231	70·455	255	77·775	279	85·095	303	92·415
160	48·800	184	56·120	208	63·440	232	70·760	256	78·080	280	85·400	304	92·720
161	49·105	185	56·425	209	63·745	233	71·065	257	78·385	281	85·705	305	93·025
162	49·410	186	56·730	210	64·050	234	71·370	258	78·790	282	86·010	306	93·330
163	49·715	187	57·035	211	64·355	235	71·675	259	79·095	283	86·315	307	93·635
164	50·020	188	57·340	212	64·660	236	71·980	260	79·400	284	86·620	308	93·940
165	50·325	189	57·645	213	64·965	237	72·285	261	79·705	285	86·925	309	94·245
166	50·630	190	57·950	214	65·270	238	72·590	262	80·010	286	87·230	310	94·550
167	50·935	191	58·255	215	65·575	239	72·895	263	80·215	287	87·535	311	94·855
168	51·240	192	58·560	216	65·880	240	73·200	264	80·520	288	87·840	312	95·160

Table CLXXXIIa.—English Feet and French Metres—*continued.*

Feet.	Metres.	Feet.	Metres.	Feet.	Metres.	Feet.	Metres.	Feet.	Metres.	Feet.	Metres.	Feet.	Metres.
313	95·465	337	102·79	361	110·11	385	117·43	409	124·75	433	132·07	457	139·39
314	95·770	338	103·09	362	110·41	386	117·73	410	125·05	434	132·37	458	139·69
315	96·075	339	103·40	363	110·72	387	118·04	411	125·36	435	132·68	459	140·00
316	96·380	340	103·70	364	111·02	388	118·34	412	125·66	436	132·98	460	140·30
317	96·685	341	104·07	365	111·33	389	118·65	413	125·97	437	133·29	461	140·61
318	96·990	342	104·31	366	111·63	390	118·95	414	126·27	438	133·59	462	140·91
319	97·295	343	104·62	367	111·94	391	119·26	415	126·58	439	133·90	463	141·22
320	97·600	344	104·92	368	112·24	392	119·56	416	126·88	440	134·20	464	141·52
321	97·905	345	105·28	369	112·55	393	119·87	417	127·19	441	134·51	465	141·85
322	98·210	346	105·53	370	112·85	394	120·17	418	127·49	442	134·81	466	142·13
323	98·515	347	105·83	371	113·16	395	120·48	419	127·80	443	135·12	467	142·44
324	98·820	348	106·13	372	113·46	396	120·78	420	128·10	444	135·42	468	142·74
325	99·125	349	106·43	373	113·77	397	121·09	421	128·41	445	135·73	469	143·05
326	99·430	350	106·75	374	114·07	398	121·39	422	121·71	446	136·03	470	143·35
327	99·735	351	107·06	375	114·38	399	121·70	423	129·02	447	136·34	471	143·66
328	100·04	352	107·36	376	114·68	400	122·00	424	129·32	448	136·64	472	143·96
329	100·35	353	107·67	377	114·99	401	122·31	425	129·63	449	136·95	473	144·27
330	100·65	354	107·97	378	115·29	402	122·61	426	129·93	450	137·25	474	144·57
331	100·96	355	108·28	379	115·60	403	122·92	427	130·24	451	137·56	475	144·88
332	101·26	356	108·58	380	115·90	404	123·22	428	130·54	452	137·86	476	145·18
333	101·57	357	108·89	381	116·21	405	123·53	429	130·85	453	138·17	477	145·49
334	101·87	358	109·19	382	116·51	406	123·83	430	131·15	454	138·47	478	145·79
335	102·18	359	109·50	383	116·82	407	124·14	431	131·46	455	138·78	479	146·10
336	102·48	360	109·80	384	117·12	408	124·44	432	131·76	456	139·08	480	146·40

Table CLXXXIII.—Decimal equivalents of Fractions of an inch.

Fractions	Decimals	Fractions.	Decimals	Fractions.	Decimals	Fractions.	Decimals
1/64	·015625	17/64	·265625	33/64	·515625	49/64	·765625
1/32	·03125	9/32	·28125	17/32	·53125	25/32	·78125
3/64	·046875	19/64	·296875	35/64	·546875	51/64	·796875
1/16	·0625	5/16	·3125	9/16	·5625	13/16	·8125
5/64	·078125	21/64	·328125	37/64	·578125	53/64	·828125
3/32	·09375	11/32	·34375	19/32	·59375	27/32	·84375
7/64	·109375	23/64	·359375	39/64	·609375	55/64	·859375
1/8	·125	3/8	·375	5/8	·625	7/8	·875
9/64	·140625	25/64	·390625	41/64	·640625	57/64	·890625
5/32	·15625	13/32	·40625	21/32	·65625	29/32	·90625
11/64	·171875	27/64	·421875	43/64	·671875	59/64	·921875
3/16	·1875	7/16	·4375	11/16	·6875	15/16	·9375
13/64	·203125	29/64	·453125	45/64	·703125	61/64	·953125
7/32	·21875	15/32	·46875	23/32	·71875	31/32	·96875
15/64	·234375	31/64	·484375	47/64	·734375	63/64	·984375
1/4	·25	1/2	·5	3/4	·75	1	1

Table CLXXXIV.—Metrical equivalents of Fractions of an inch, &c.

Fractions of an inch.	Millimetres.	Fractions of an inch.	Millimetres.	Inches.	Millimetres.	Inches.	Millimetres.
1/32	0·7937	17/32	13·4935	1	25·3995	17	431·7922
1/16	1·5875	9/16	14·2872	2	50·7991	18	457·1917
3/32	2·3812	19/32	15·0810	3	76·1986	19	482·5913
1/8	3·1749	5/8	15·8747	4	101·5982	20	507·9908
5/32	3·9687	21/32	16·6684	5	126·9977	21	533·3904
3/16	4·7624	11/16	17·4622	6	152·3972	22	558·7899
7/32	5·5561	23/32	18·2559	7	177·7968	23	584·1894
1/4	6·3499	3/4	19·0496	8	203·1963	24	609·5890
9/32	7·1436	25/32	19·8434	9	228·5959	25	634·9885
5/16	7·9374	13/16	20·6371	10	253·9954	26	660·3881
11/32	8·7311	27/32	21·4309	11	279·3950	27	685·7876
3/8	9·5248	7/8	22·2246	12	304·7945	28	711·1872
13/32	10·3186	29/32	23·0183	13	330·1940	29	736·5867
7/16	11·1123	15/16	23·8121	14	355·5936	30	761·9862
15/32	11·9060	31/32	24·6058	15	380·9931	31	787·3858
1/2	12·6998			16	406·3926	32	812·7853

Table CLXXXV.—Square Feet and Square Metres.

Square Feet.	Square Metres.	Square Feet.	Square Metres.	Square Feet.	Square Metres.	Square Feet.	Square Metres.
1	·0929	26	2·4154	51	4·7379	76	7·0604
2	·1858	27	2·5083	52	4·8308	77	7·1533
3	·2787	28	2·6012	53	4·9237	78	7·2462
4	·3716	29	2·6941	54	5·0166	79	7·3391
5	·4645	30	2·7870	55	5·1095	80	7·4320
6	·5574	31	2·8799	56	5·2024	81	7·5249
7	·6503	32	2·9728	57	5·2953	82	7·6178
8	·7432	33	3·0657	58	5·3882	83	7·7107
9	·8361	34	3·1586	59	5·4811	84	7·8036
10	·9290	35	3·2515	60	5·5740	85	7·8965
11	1·0219	36	3·3444	61	5·6669	86	7·9894
12	1·1148	37	3·4373	62	5·7598	87	8·0823
13	1·2077	38	3·5302	63	5·8527	88	8·1752
14	1·3006	39	3·6231	64	5·9456	89	8·2681
15	1·3935	40	3·7160	65	6·0385	90	8·3610
16	1·4864	41	3·8089	66	6·1314	91	8·4539
17	1·5793	42	3·9018	67	6·2243	92	8·5468
18	1·6722	43	3·9947	68	6·3172	93	8·6397
19	1·7651	44	4·0876	69	6·4101	94	8·7326
20	1·8580	45	4·1805	70	6·5030	95	8·8255
21	1·9509	46	4·2734	71	6·5969	96	8·9184
22	2·0438	47	4·3663	72	6·6888	97	9·0113
23	2·1367	48	4·4592	73	6·7817	98	9·1042
24	2·2296	49	4·5521	74	6·8746	99	9·1971
25	2·3225	50	4·6450	75	6·9675	100	9·2900

The above Table can, of course, be used for hundreds and thousands of feet, or for hundredths and thousandths of feet, by altering the position of the decimal point: *e.g.*,—50 square feet = 4·645 square metres, and 5000 square feet = 464·5 square metres; also ·5 square foot = ·04645 square metre.

Table CLXXXVI.—Square Metres and Square Feet.

Square Metres.	Square Feet.	Square Metres.	Square Feet.	Square Metres.	Square Feet.	Square Metres.	Square Feet
1	10·764	26	279·872	51	548·979	76	818·087
2	21·529	27	290·636	52	559·744	77	828·851
3	32·293	28	301·400	53	570·508	78	839·615
4	43·057	29	312·165	54	581·272	79	850·380
5	53·822	30	322·929	55	592·036	80	861·144
6	64·586	31	333·693	56	602·801	81	871·908
7	75·350	32	344·458	57	613·565	82	882·673
8	86·114	33	355·222	58	624·329	83	893·437
9	96·879	34	365·986	59	635·094	84	904·201
10	107·643	35	376·750	60	645·858	85	914·966
11	118·407	36	387·515	61	656·622	86	925·730
12	129·172	37	398·279	62	667·387	87	936·494
13	139·936	38	409·043	63	678·151	88	947·258
14	150·700	39	419·808	64	688·915	89	958·023
15	161·464	40	430·572	65	699·680	90	968·787
16	172·229	41	441·336	66	710·444	91	979·551
17	182·993	42	452·101	67	721·208	92	990·316
18	193·757	43	462·865	68	731·972	93	1001·080
19	204·522	44	473·629	69	742·737	94	1011·844
20	215·286	45	484·394	70	753·501	95	1022·608
21	226·050	46	495·158	71	764·265	96	1033·373
22	236·815	47	505·922	72	775·030	97	1044·137
23	247·579	48	516·686	73	785·794	98	1054·901
24	258·343	49	527·451	74	796·558	99	1065·666
25	269·108	50	538·215	75	807·322	100	1076·430

The above Table can, of course, be used for hundreds and thousands of metres, or for hundredths and thousandths of metres, by altering the position of the decimal point: *e.g.*,—50 square metres = 538·215 square feet, and 5000 square metres = 53821·5 square feet; also ·5 square metre = 5·38215 square feet.

Table CLXXXVII.—English Weights and Metric Equivalents.

Lbs.	Kilogrammes.	Lbs.	Kilogrammes.	Lbs.	Kilogrammes.
1	·4536	42	19·0509	83	37·6482
2	·9072	43	19·5045	84	38·1018
3	1·3608	44	19·9581	85	38·5554
4	1·8144	45	20·4117	86	39·0089
5	2·2680	46	20·8653	87	39·4625
6	2·7216	47	21·3189	88	39·9161
7	3·1752	48	21·7724	89	40·3697
8	3·6287	49	22·2260	90	40·8233
9	4·0823	50	22·6796	91	41·2769
10	4·5359	51	23·1332	92	41·7305
11	4·9895	52	23·5868	93	42·1841
12	5·4431	53	24·0404	94	42·6377
13	5·8967	54	24·4940	95	43·0913
14	6·3503	55	24·9476	96	43·5449
15	6·8039	56	25·4012	97	43·9985
16	7·2575	57	25·8548	98	44·4521
17	7·7111	58	26·3084	99	44·9057
18	8·1647	59	26·7619	100	45·3593
19	8·6182	60	27·2155	101	45·8128
20	9·0718	61	27·6691	102	46·2664
21	9·5254	62	28·1227	103	46·7200
22	9·9790	63	28·5763	104	47·1736
23	10·4326	64	29·0299	105	47·6272
24	10·8862	65	29·4835	106	48·0808
25	11·3398	66	29·9371	107	48·5344
26	11·7934	67	30·3907	108	48·9880
27	12·2470	68	30·8443	109	49·4416
28	12·7006	69	31·2979	110	49·8952
29	13·1542	70	31·7515	111	50·3488
30	13·6078	71	32·3051	112	50·8024
31	14·0614	72	32·6587	200	90·7185
32	14·5149	73	33·1123	300	136·0778
33	14·9685	74	33·5658	400	181·4370
34	15·4221	75	34·0194	500	226·7963
35	15·8757	76	34·4730	600	272·1556
36	16·3292	77	34·9266	700	317·5148
37	16·7293	78	35·3802	800	362·8741
38	17·2365	79	35·8338	900	408·2334
39	17·6901	80	36·2874	1000	453·5926
40	18·1437	81	36·7410	2000	907·1853
41	18·5973	82	37·1946	2240	1016·0475

27 lbs. = 12·247 kilos. ; 2·7 lbs. = 1·2247 kilos. ; 2700 lbs. = 1224·7 kilos., &c.

Table CLXXXVIIa.—Metric Weights and English Equivalents.

Kilo-grams.	Lbs.	Kilo-grams.	Lbs.	Kilo-grams.	Lbs.
1	2·2046	38	83·7756	75	165·3466
2	4·4092	39	85·9802	76	167·5512
3	6·6139	40	88·1848	77	169·7559
4	8·8185	41	90·3895	78	171·9605
5	11·0231	42	92·5941	79	174·1651
6	13·2277	43	94·7987	80	176·3697
7	15·4324	44	97·0034	81	178·5743
8	17·6370	45	99·2079	82	180·7789
9	19·8416	46	101·4126	83	182·9836
10	22·0462	47	103·6172	84	185·1882
11	24·2508	48	105·8218	85	187·3928
12	26·4554	49	108·0264	86	189·5974
13	28·6601	50	110·2311	87	191·8020
14	30·8647	51	112·4357	88	194·0067
15	33·0693	52	114·6403	89	196·2113
16	35·2739	53	116·8499	90	198·4159
17	37·4786	54	119·0495	91	200·6205
18	39·6832	55	121·2542	92	202·8251
19	41·8878	56	123·4588	93	205·0298
20	44·0924	57	125·6634	94	207·2344
21	46·2970	58	127·8680	95	209·4390
22	48·5017	59	130·0727	96	211·6431
23	50·7063	60	132·2773	97	213·8482
24	52·9109	61	134·4819	98	216·0529
25	55·1155	62	136·6865	99	218·2575
26	57·3202	63	138·8911	100	220·4621
27	59·5248	64	141·0958	200	440·9243
28	61·7294	65	143·3004	300	661·3864
29	63·9340	66	145·5050	400	881·8485
30	66·1386	67	147·7096	500	1102·3106
31	68·3433	68	149·9142	600	1322·7728
32	70·5479	69	152·1189	700	1543·2349
33	72·7525	70	154·3235	800	1763·6970
34	74·9571	71	156·5281	900	1984·1591
35	77·1617	72	158·7327	1000	2204·6213
36	79·3664	73	160·9374	1016	2239·8952
37	81·5709	74	163·1419		

58 kilos. = 127·868 lbs. ; 5·8 kilos. = 12·7868 lbs. ; 5800 kilos. = 12786·8 lbs., &c.

Table CLXXXVIII.—Pounds per square inch and Kilogrammes per square centimetre.

Lbs. per sq. inch.	Kilos. per sq. cm.	Lbs. per sq. inch.	Kilos. per sq. cm.	Lbs. per sq. inch.	Kilos. per sq. cm.	Lbs. per sq. inch.	Kilos. per sq. cm.	Lbs. per sq. inch.	Kilos. per sq. cm.
1	·0703	35	2·460	69	4·850	103	7·241	137	9·632
2	·1406	36	2·530	70	4·921	104	7·312	138	9·702
3	·2109	37	2·601	71	4·991	105	7·382	139	9·772
4	·2812	38	2·671	72	5·061	106	7·452	140	9·843
5	·3515	39	2·741	73	5·131	107	7·522	141	9·913
6	·4218	40	2·812	74	5·202	108	7·593	142	9·983
7	·4921	41	2·882	75	5·272	109	7·663	143	10·054
8	·5624	42	2·952	76	5·342	110	7·733	144	10·124
9	·6327	43	3·022	77	5·413	111	7·804	145	10·194
10	·7030	44	3·093	78	5·483	112	7·874	146	10·264
11	·7733	45	3·163	79	5·553	113	7·944	147	10·335
12	·8436	46	3·233	80	5·624	114	8·015	148	10·405
13	·9140	47	3·304	81	5·694	115	8·085	149	10·475
14	·9843	48	3·374	82	5·764	116	8·155	150	10·546
15	1·0546	49	3·444	83	5·834	117	8·226	155	10·897
16	1·1248	50	3·515	84	5·905	118	8·296	160	11·249
17	1·1952	51	3·585	85	5·975	119	8·366	165	11·600
18	1·265	52	3·655	86	6·045	120	8·436	170	11·952
19	1·335	53	3·725	87	6·116	121	8·507	175	12·303
20	1·406	54	3·796	88	6·186	122	8·577	180	12·655
21	1·476	55	3·866	89	6·256	123	8·647	185	13·006
22	1·546	56	3·936	90	6·327	124	8·718	190	13·358
23	1·616	57	4·007	91	6·397	125	8·788	195	13·710
24	1·687	58	4·077	92	6·467	126	8·858	200	14·061
25	1·757	59	4·147	93	6·537	127	8·929	210	14·76
26	1·827	60	4·218	94	6·608	128	8·999	220	15·46
27	1·898	61	4·288	95	6·678	129	9·069	230	16·16
28	1·968	62	4·358	96	6·748	130	9·140	240	16·87
29	2·038	63	4·428	97	6·819	131	9·210	250	17·57
30	2·109	64	4·499	98	6·889	132	9·280	260	18·27
31	2·179	65	4·569	99	6·959	133	9·350	270	18·98
32	2·249	66	4·639	100	7·030	134	9·421	280	19·68
33	2·319	67	4·710	101	7·101	135	9·491	290	20·38
34	2·390	68	4·780	102	7·171	136	9·561	300	21·09

Table CLXXXIX.—Kilogrammes per square centimetre and Pounds per square inch.

Kilos. per sq. cm.	Lbs. per square inch.	Kilos. per sq. cm.	Lbs. per square inch.	Kilos. per sq. cm.	Lbs. per square inch.	Kilos. per sq. cm.	Lbs. per square inch.
·1	1·422	3·1	44·091	6·1	86·761	9·1	129·431
·2	2·844	3·2	45·514	6·2	88·183	9·2	130·853
·3	4·266	3·3	46·936	6·3	89·606	9·3	132·275
·4	5·689	3·4	48·358	6·4	91·028	9·4	133·698
·5	7·111	3·5	49·781	6·5	92·450	9·5	135·120
·6	8·533	3·6	51·203	6·6	93·873	9·6	136·542
·7	9·956	3·7	52·625	6·7	95·295	9·7	137·965
·8	11·378	3·8	54·048	6·8	96·717	9·8	139·387
·9	12·800	3·9	55·470	6·9	98·140	9·9	140·809
1·0	14·223	4·0	56·892	7·0	99·562	10·0	142·232
1·1	15·645	4·1	58·315	7·1	100·984	10·5	149·343
1·2	17·067	4·2	59·737	7·2	102·407	11·0	156·455
1·3	18·490	4·3	61·159	7·3	103·829	11·5	163·566
1·4	19·912	4·4	62·582	7·4	105·251	12·0	170·678
1·5	21·334	4·5	64·004	7·5	106·674	12·5	177·790
1·6	22·757	4·6	65·426	7·6	108·096	13·0	184·901
1·7	24·179	4·7	66·849	7·7	109·518	13·5	192·013
1·8	25·601	4·8	68·271	7·8	110·940	14·0	199·124
1·9	27·024	4·9	69·693	7·9	112·363	14·5	206·236
2·0	28·446	5·0	71·116	8·0	113·785	15·0	213·348
2·1	29·868	5·1	72·538	8·1	115·207	15·5	220·459
2·2	31·291	5·2	73·960	8·2	116·630	16·0	227·571
2·3	32·713	5·3	75·382	8·3	118·052	16·5	234·682
2·4	34·135	5·4	76·805	8·4	119·474	17·0	241·794
2·5	35·558	5·5	78·227	8·5	120·897	17·5	248·906
2·6	36·980	5·6	79·649	8·6	122·319	18·0	256·017
2·7	38·402	5·7	81·072	8·7	123·741	18·5	263·129
2·8	39·824	5·8	82·494	8·8	125·164	19·0	270·240
2·9	41·247	5·9	83·916	8·9	126·586	19·5	277·352
3·0	42·669	6·0	85·339	9·0	128·008	20·0	284·464

Table CXC.—Kilogrammes per square millimetre (or centimetre) and Tons per square inch.*

Kilos. per mm.²	Tons per sq. in.	Kilos. per mm.²	Tons per sq. in.	Kilos. per mm.²	Tons per sq. in.	Kilos. per mm.²	Tons per sq. in.
·5	·318	17·5	11·112	34·5	21·908	51·5	32·702
1·0	·635	18·0	11·430	35·0	22·225	52·0	33·020
1·5	·952	18·5	11·748	35·5	22·542	52·5	33·338
2·0	1·270	19·0	12·065	36·0	22·860	53·0	33·655
2·5	1·588	19·5	12·382	36·5	23·178	53·5	33·972
3·0	1·905	20·0	12·700	37·0	23·495	54·0	34·290
3·5	2·222	20·5	13·018	37·5	23·812	54·5	34·608
4·0	2·540	21·0	13·335	38·0	24·130	55·0	34·925
4·5	2·858	21·5	13·652	38·5	24·448	55·5	35·282
5·0	3·175	22·0	13·970	39·0	24·765	56·0	35·560
5·5	3·492	22·5	14·288	39·5	25·082	56·5	35·878
6·0	3·810	23·0	14·605	40·0	25·400	57·0	36·185
6·5	4·128	23·5	14·922	40·5	25·718	57·5	36·502
7·0	4·445	24·0	15·240	41·0	26·035	58·0	36·830
7·5	4·762	24·5	15·558	41·5	26·352	58·5	37·148
8·0	5·080	25·0	15·875	42·0	26·670	59·0	37·465
8·5	5·398	25·5	16·192	42·5	26·988	59·5	37·472
9·0	5·715	26·0	16·510	43·0	27·305	60·0	38·100
9·5	6·032	26·5	16·828	43·5	27·622	60·5	38·418
10·0	6·350	27·0	17·145	44·0	27·940	61·0	38·785
10·5	6·668	27·5	17·462	44·5	28·258	61·5	39·052
11·0	6·985	28·0	17·780	45·0	28·575	62·0	39·370
11·5	7·302	28·5	18·098	45·5	28·892	62·5	39·688
12·0	7·620	29·0	18·415	46·0	29·210	63·0	40·005
12·5	7·938	29·5	18·732	46·5	29·528	63·5	40·322
13·0	8·255	30·0	19·050	47·0	29·845	64·0	40·640
13·5	8·572	30·5	19·368	47·5	30·162	64·5	40·958
14·0	8·890	31·0	19·685	48·0	30·480	65·0	41·275
14·5	9·208	31·5	20·002	48·5	30·798	65·5	41·592
15·0	9·525	32·0	20·320	49·0	31·115	66·0	41·910
15·5	9·842	32·5	20·638	49·5	31·432	66·5	42·238
16·0	10·160	33·0	20·955	50·0	31·750	67·0	42·545
16·5	10·478	33·5	21·272	50·5	32·068	67·5	42·852
17·0	10·795	34·0	21·590	51·0	32·385	68·0	43·170

* 1 kilo. per sq. millimetre = 100 kilos. per sq. centimetre, etc.

Table CXCI.—Areas of Segments of Circles.

To find the area of any segment of a circle,—Divide the versed sine or height of the segment (V) by the diameter of the circle of which it is a part (D), and multiply the square of the diameter by the value of x (see Table) corresponding to the value of $\frac{V}{D}$ obtained; that is,—

$$\text{Area of Segment} = \text{Diameter}^2 \times x.$$

$\frac{V}{D}$	x	$\frac{V}{D}$	x	$\frac{V}{D}$	x	$\frac{V}{D}$	x
·001	·000042	·038	·009763	·075	·026761	·112	·048262
·002	·000119	·039	·010148	·076	·027289	·113	·048894
·003	·000219	·040	·010537	·077	·027821	·114	·049528
·004	·000337	·041	·010931	·078	·028356	·115	·050165
·005	·000470	·042	·011330	·079	·028894	·116	·050804
·006	·000618	·043	·011734	·080	·029435	·117	·051446
·007	·000779	·044	·012142	·081	·029979	·118	·052090
·008	·000951	·045	·012554	·082	·030526	·119	·052736
·009	·001135	·046	·012971	·083	·031076	·120	·053385
·010	·001329	·047	·013392	·084	·031629	·121	·054036
·011	·001533	·048	·013818	·085	·032186	·122	·054689
·012	·001746	·049	·014247	·086	·032745	·123	·055345
·013	·001968	·050	·014681	·087	·033307	·124	·056003
·014	·002199	·051	·015119	·088	·033872	·125	·056663
·015	·002438	·052	·015561	·089	·034441	·126	·057326
·016	·002685	·053	·016007	·090	·035011	·127	·057991
·017	·002940	·054	·016457	·091	·035585	·128	·058658
·018	·003202	·055	·016911	·092	·036162	·129	·059327
·019	·003471	·056	·017369	·093	·036741	·130	·059999
·020	·003748	·057	·017831	·094	·037323	·131	·060672
·021	·004031	·058	·018296	·095	·037909	·132	·061348
·022	·004322	·059	·018766	·096	·038496	·133	·062026
·023	·004618	·060	·019239	·097	·039087	·134	·062707
·024	·004921	·061	·019716	·098	·039680	·135	·063389
·025	·005230	·062	·020196	·099	·040276	·136	·064074
·026	·005546	·063	·020680	·100	·040875	·137	·064760
·027	·005867	·064	·021168	·101	·041476	·138	·065449
·028	·006194	·065	·021659	·102	·042080	·139	·066140
·029	·006527	·066	·022154	·103	·042687	·140	·066833
·030	·006865	·067	·022652	·104	·043296	·141	·067528
·031	·007209	·068	·023154	·105	·043908	·142	·068225
·032	·007558	·069	·023659	·106	·044522	·143	·068924
·033	·007913	·070	·024168	·107	·045139	·144	·069625
·034	·008273	·071	·024680	·108	·045759	·145	·070328
·035	·008638	·072	·025195	·109	·046381	·146	·071033
·036	·009008	·073	·025714	·110	·047005	·147	·071741
·037	·009383	·074	·026236	·111	·047632	·148	·072450

Table CXCI.—Areas of Segments of Circles—*continued.*

$\frac{V}{D}$	s	$\frac{V}{D}$	s	$\frac{V}{D}$	s	$\frac{V}{D}$	s
·149	·078161	·193	·106261	·237	·142387	·281	·180918
·150	·078874	·194	·107051	·238	·143238	·282	·181817
·151	·074589	·195	·107842	·239	·144091	·283	·182718
·152	·075306	·196	·108636	·240	·144944	·284	·183619
·153	·076026	·197	·109430	·241	·145799	·285	·184521
·154	·076747	·198	·110226	·242	·146655	·286	·185425
·155	·077469	·199	·111024	·243	·147512	·287	·186329
·156	·078194	·200	·111823	·244	·148371	·288	·187234
·157	·078921	·201	·112624	·245	·149230	·289	·188140
·158	·079649	·202	·113426	·246	·150091	·290	·189047
·159	·080380	·203	·114230	·247	·150953	·291	·189955
·160	·081112	·204	·115035	·248	·151816	·292	·190864
·161	·081846	·205	·115842	·249	·152680	·293	·191775
·162	·082582	·206	·116650	·250	·153546	·294	·192684
·163	·083320	·207	·117460	·251	·154412	·295	·193596
·164	·084059	·208	·118271	·252	·155280	·296	·194509
·165	·084801	·209	·119083	·253	·156149	·297	·195422
·166	·085544	·210	·119897	·254	·157019	·298	·196337
·167	·086289	·211	·120712	·255	·157890	·299	·197252
·168	·087036	·212	·121529	·256	·158762	·300	·198168
·169	·087785	·213	·122347	·257	·159636	·301	·199085
·170	·088535	·214	·123167	·258	·160510	·302	·200003
·171	·089287	·215	·123988	·259	·161386	·303	·200922
·172	·090041	·216	·124810	·260	·162263	·304	·201841
·173	·090797	·217	·125634	·261	·163140	·305	·202761
·174	·091554	·218	·126459	·262	·164019	·306	·203683
·175	·092313	·219	·127285	·263	·164899	·307	·204605
·176	·093074	·220	·128113	·264	·165780	·308	·205527
·177	·093836	·221	·128942	·265	·166663	·309	·206451
·178	·094601	·222	·129773	·266	·167546	·310	·207376
·179	·095366	·223	·130605	·267	·168430	·311	·208301
·180	·096134	·224	·131438	·268	·169315	·312	·209227
·181	·096903	·225	·132272	·269	·170202	·313	·210154
·182	·097674	·226	·133108	·270	·171089	·314	·211082
·183	·098447	·227	·133945	·271	·171978	·315	·212011
·184	·099221	·228	·134784	·272	·172867	·316	·212940
·185	·099997	·229	·135624	·273	·173758	·317	·213871
·186	·100774	·230	·136465	·274	·174649	·318	·214802
·187	·101553	·231	·137307	·275	·175542	·319	·215733
·188	·102334	·232	·138150	·276	·176435	·320	·216666
·189	·103116	·233	·138995	·277	·177330	·321	·217599
·190	·103900	·234	·139841	·278	·178225	·322	·218533
·191	·104685	·235	·140688	·279	·179122	·323	·219468
·192	·105472	·236	·141537	·280	·180019	·324	·220404

Table CXCI.—Areas of Segments of Circles—*continued.*

$\frac{v}{D}$	z	$\frac{v}{D}$	z	$\frac{v}{D}$	z	$\frac{v}{D}$	z
·325	·221340	·369	·263213	·413	·306140	·457	·349752
·326	·222277	·370	·264178	·414	·307125	·458	·350748
·327	·223215	·371	·265144	·415	·308110	·459	·351745
·328	·224154	·372	·266111	·416	·309095	·460	·352742
·329	·225093	·373	·267078	·417	·310081	·461	·353739
·330	·226033	·374	·268045	·418	·311068	·462	·354736
·331	·226974	·375	·269013	·419	·312054	·463	·355732
·332	·227915	·376	·269982	·420	·313041	·464	·356730
·333	·228858	·377	·270951	·421	·314029	·465	·357727
·334	·229801	·378	·271920	·422	·315016	·466	·358725
·335	·230745	·379	·272890	·423	·316004	·467	·359723
·336	·231689	·380	·273861	·424	·316992	·468	·360721
·337	·232634	·381	·274832	·425	·317981	·469	·361719
·338	·233580	·382	·275803	·426	·318970	·470	·362717
·339	·234526	·383	·276775	·427	·319959	·471	·363715
·340	·235473	·384	·277748	·428	·320948	·472	·364713
·341	·236421	·385	·278721	·429	·321938	·473	·365712
·342	·237369	·386	·279694	·430	·322928	·474	·366710
·343	·238318	·387	·280668	·431	·323918	·475	·367709
·344	·239268	·388	·281642	·432	·324909	·476	·368708
·345	·240218	·389	·282617	·433	·325900	·477	·369707
·346	·241169	·390	·283592	·434	·326892	·478	·370706
·347	·242121	·391	·284568	·435	·327882	·479	·371705
·348	·243074	·392	·285544	·436	·328874	·480	·372704
·349	·244026	·393	·286521	·437	·329866	·481	·373703
·350	·244980	·394	·287498	·438	·330858	·482	·374702
·351	·245934	·395	·288476	·439	·331850	·483	·375702
·352	·246889	·396	·289453	·440	·332843	·484	·376702
·353	·247845	·397	·290432	·441	·333836	·485	·377701
·354	·248801	·398	·291411	·442	·334829	·486	·378701
·355	·249757	·399	·292390	·443	·335822	·487	·379700
·356	·250715	·400	·293369	·444	·336816	·488	·380700
·357	·251673	·401	·294349	·445	·337810	·489	·381699
·358	·252631	·402	·295330	·446	·338804	·490	·382699
·359	·253590	·403	·296311	·447	·339798	·491	·383699
·360	·254550	·404	·297292	·448	·340793	·492	·384699
·361	·255510	·405	·298273	·449	·341787	·493	·385699
·362	·256471	·406	·299255	·450	·342782	·494	·386699
·363	·257433	·407	·300238	·451	·343777	·495	·387699
·364	·258395	·408	·301220	·452	·344772	·496	·388699
·365	·259357	·409	·302203	·453	·345768	·497	·389699
·366	·260320	·410	·303187	·454	·346764	·498	·390699
·367	·261284	·411	·304171	·455	·347759	·499	·391699
·368	·262248	·412	·305155	·456	·348755	·500	·392699

Table CXCII.—Areas of Circles.

Diameters.	Areas.								Diameters.
	·0	·⅛	·¼	·⅜	·½	·⅝	·¾	·⅞	
0	·0	·01227	·0491	·1104	·1963	·3068	·4418	·6013	0
1	·78540	·99402	1·2272	1·4849	1·7671	2·0739	2·4053	2·7612	1
2	3·1416	3·5466	3·9761	4·4301	4·9087	5·4119	5·9396	6·4918	2
3	7·0686	7·6699	8·2958	8·9462	9·6211	10·3206	11·0447	11·7933	3
4	12·5664	13·3641	14·1863	15·0330	15·9043	16·8002	17·7206	18·6655	4
5	19·6350	20·6290	21·6476	22·6907	23·7583	24·8505	25·9673	27·1086	5
6	28·2744	29·4648	30·6797	31·9191	33·1831	34·4717	35·7848	37·1224	6
7	38·4846	39·8713	41·2826	42·7184	44·1787	45·6636	47·1731	48·7071	7
8	50·2656	51·8487	53·4563	55·0884	56·7451	58·4264	60·1322	61·8625	8
9	63·6174	65·3968	67·2008	69·0293	70·8823	72·7599	74·6621	76·5888	9
10	78·5400	80·5158	82·5161	84·5409	86·5903	88·6643	90·7628	92·8858	10
11	95·0334	97·2055	99·5022	101·6234	103·8691	106·1394	108·4343	110·7537	11
12	113·098	115·466	117·859	120·277	122·719	125·185	127·677	130·192	12
13	132·738	135·297	137·887	140·501	143·139	145·802	148·490	151·202	13
14	153·938	156·700	159·485	162·296	165·130	167·990	170·874	173·782	14
15	176·715	179·673	182·655	185·661	188·692	191·748	194·828	197·933	15
16	201·062	204·216	207·395	210·598	213·825	217·077	220·354	223·655	16
17	226·981	230·331	233·706	237·105	240·529	243·977	247·450	250·948	17
18	254·470	258·016	261·587	265·183	268·803	272·448	276·117	279·811	18
19	283·529	287·272	291·040	294·832	298·648	302·489	306·355	310·245	19
20	314·160	318·099	322·063	326·051	330·064	334·102	338·164	342·250	20
21	346·361	350·497	354·657	358·842	363·051	367·285	371·543	375·826	21
22	380·134	384·466	388·822	393·203	397·609	402·038	406·494	410·973	22

Table CXCII.—Areas of Circles—*continued.*

Diameters.	Areas.								Diameters.
	·0	·⅛	·¼	·⅜	·½	·⅝	·¾	·⅞	
23	415·477	420·004	424·558	429·135	433·737	438·364	443·015	447·690	23
24	452·390	457·115	461·864	466·638	471·436	476·259	481·107	485·979	24
25	490·875	495·796	500·742	505·712	510·706	515·726	520·769	525·838	25
26	530·930	536·048	541·190	546·356	551·547	556·763	562·003	567·267	26
27	572·557	577·870	583·209	588·571	593·959	599·371	604·807	610·268	27
28	615·754	621·264	626·798	632·357	637·941	643·549	649·182	654·840	28
29	660·521	666·228	671·959	677·714	683·494	689·299	695·128	700·982	29
30	706·860	712·763	718·690	724·642	730·618	736·619	742·645	748·695	30
31	754·769	760·869	766·992	773·140	779·313	785·510	791·732	797·979	31
32	804·250	810·545	816·865	823·210	829·579	835·972	842·391	848·833	32
33	855·301	861·792	868·309	874·850	881·415	888·005	894·620	901·259	33
34	907·922	914·611	921·323	928·061	934·822	941·609	948·420	955·255	34
35	962·115	969·000	975·909	982·842	989·800	996·783	1003·790	1010·822	35
36	1017·878	1024·960	1032·065	1039·195	1046·349	1053·528	1060·732	1067·960	36
37	1075·213	1082·490	1089·792	1097·118	1104·469	1111·844	1119·244	1126·669	37
38	1134·118	1141·591	1149·089	1156·612	1164·159	1171·731	1179·327	1186·948	38
39	1194·593	1202·263	1209·958	1217·677	1225·420	1233·188	1240·981	1248·798	39
40	1256·64	1264·51	1272·40	1280·31	1288·25	1296·22	1304·21	1312·22	40
41	1320·26	1328·32	1336·41	1344·52	1352·66	1360·82	1369·00	1377·21	41
42	1385·45	1393·70	1401·99	1410·30	1418·63	1426·99	1435·37	1443·77	42
43	1452·20	1460·66	1469·14	1477·64	1486·17	1494·73	1503·30	1511·91	43
44	1520·53	1529·19	1537·86	1546·56	1555·29	1564·04	1572·81	1581·61	44

Table CXCII.—Areas of Circles—*continued.*

Diameters.	Areas.								Diameters.
	·0	·⅛	·¼	·⅜	·½	·⅝	·¾	·⅞	
45	1590·43	1599·28	1608·16	1617·05	1625·97	1634·92	1643·89	1652·89	45
46	1661·91	1670·95	1680·02	1689·11	1698·23	1707·37	1716·54	1725·73	46
47	1734·95	1744·19	1753·45	1762·74	1772·06	1781·40	1790·76	1800·15	47
48	1809·56	1819·00	1828·46	1837·95	1847·46	1856·99	1866·55	1876·14	48
49	1885·75	1895·38	1905·04	1914·72	1924·43	1934·16	1943·91	1953·69	49
50	1963·50	1973·33	1983·18	1993·06	2002·97	2012·89	2022·85	2032·82	50
51	2042·83	2052·85	2062·90	2072·98	2083·08	2093·20	2103·35	2113·52	51
52	2123·72	2133·94	2144·19	2154·46	2164·76	2175·08	2185·42	2195·79	52
53	2206·19	2216·61	2227·05	2237·52	2248·01	2258·53	2269·07	2279·64	53
54	2290·23	2300·84	2311·48	2322·15	2332·83	2343·55	2354·29	2365·05	54
55	2375·83	2386·65	2397·48	2408·34	2419·23	2430·14	2441·07	2452·03	55
56	2463·01	2474·02	2485·05	2496·11	2507·19	2518·30	2529·43	2540·58	56
57	2551·76	2562·97	2574·20	2585·45	2596·73	2608·03	2619·36	2630·71	57
58	2642·09	2653·49	2664·91	2676·36	2687·84	2699·33	2710·86	2722·41	58
59	2733·98	2745·57	2757·20	2768·84	2780·51	2792·21	2803·93	2815·67	59
60	2827·44	2839·23	2851·05	2862·89	2874·76	2886·65	2898·57	2910·51	60
61	2922·47	2934·46	2946·48	2958·52	2970·58	2982·67	2994·78	3006·92	61
62	3019·08	3031·26	3043·47	3055·71	3067·97	3080·25	3092·56	3104·89	62
63	3117·25	3129·64	3142·04	3154·47	3166·93	3179·41	3191·91	3204·44	63
64	3217·00	3229·58	3242·18	3254·81	3267·46	3280·14	3292·84	3305·56	64
65	3318·31	3331·09	3343·89	3356·71	3369·56	3382·44	3395·33	3408·26	65
66	3421·20	3434·17	3447·17	3460·19	3473·24	3486·30	3499·40	3512·52	66

Table CXCII.—Areas of Circles—*continued.*

Diameters.	Areas.								Diameters.
	·0	·⅛	·¼	·⅜	·½	·⅝	·¾	·⅞	
67	3525·66	3538·83	3552·02	3565·24	3578·48	3591·74	3605·04	3618·35	67
68	3631·69	3645·05	3658·44	3671·86	3685·29	3698·76	3712·24	3725·75	68
69	3739·29	3752·85	3766·43	3780·04	3793·68	3807·34	3821·02	3834·73	69
70	3848·46	3862·22	3876·00	3889·80	3903·63	3917·49	3931·37	3945·27	70
71	3959·20	3973·15	3987·13	4001·13	4015·16	4029·21	4043·29	4057·39	71
72	4071·51	4085·66	4099·84	4114·04	4128·26	4142·51	4156·78	4171·08	72
73	4185·40	4199·74	4214·11	4228·51	4242·93	4257·37	4271·84	4286·33	73
74	4300·85	4315·39	4329·96	4344·55	4359·17	4373·81	4388·47	4403·16	74
75	4417·87	4432·61	4447·38	4462·16	4476·98	4491·81	4506·67	4521·56	75
76	4536·47	4551·41	4566·36	4581·35	4596·36	4611·39	4626·45	4641·53	76
77	4656·64	4671·77	4686·92	4702·10	4717·31	4732·54	4747·79	4763·07	77
78	4778·37	4793·70	4809·05	4824·43	4839·83	4855·26	4870·71	4886·18	78
79	4901·68	4917·21	4932·75	4948·33	4963·92	4979·55	4995·19	5010·86	79
80	5026·56	5042·28	5058·03	5073·79	5089·59	5105·41	5121·25	5137·12	80
81	5153·01	5168·93	5184·87	5200·83	5216·82	5232·84	5248·88	5264·94	81
82	5281·03	5297·14	5313·28	5329·44	5345·63	5361·84	5378·08	5394·34	82
83	5410·62	5426·93	5443·26	5459·62	5476·01	5492·41	5508·84	5525·30	83
84	5541·78	5558·29	5574·82	5591·37	5607·95	5624·56	5641·18	5657·84	84
85	5674·51	5691·22	5707·94	5724·69	5741·47	5758·27	5775·10	5791·94	85
86	5808·82	5825·72	5842·64	5859·59	5876·56	5893·55	5910·58	5927·62	86
87	5944·69	5961·79	5978·91	5996·05	6013·22	6030·41	6047·63	6064·87	87
88	6082·14	6099·43	6116·74	6134·08	6151·45	6168·84	6186·25	6203·69	88

Table CXCII.—Areas of Circles—*continued.*

Diameters.	Areas.								Diameters
	·0	·⅛	·¼	·⅜	·½	·⅝	·¾	·⅞	
89	6221·15	6238·64	6256·15	6273·69	6291·25	6308·84	6326·45	6344·08	89
90	6361·74	6379·42	6397·13	6414·86	6432·62	6450·40	6468·21	6486·04	90
91	6503·90	6521·78	6539·68	6557·61	6575·56	6593·54	6611·55	6629·57	91
92	6647·63	6665·70	6683·80	6701·93	6720·08	6738·25	6756·45	6774·68	92
93	6792·92	6811·20	6829·49	6847·82	6866·16	6884·53	6902·93	6921·35	93
94	6939·79	6958·26	6976·76	6995·28	7013·82	7032·39	7050·98	7069·59	94
95	7088·24	7106·90	7125·59	7144·31	7163·04	7181·81	7200·60	7219·41	95
96	7238·25	7257·11	7275·99	7294·91	7313·84	7332·80	7351·79	7370·79	96
97	7389·83	7408·89	7427·97	7447·08	7466·21	7485·37	7504·55	7523·75	97
98	7542·98	7562·24	7581·52	7600·82	7620·15	7639·50	7658·88	7678·28	98
99	7697·71	7717·16	7736·63	7756·13	7775·66	7795·21	7814·78	7834·38	99
100	7854·00	7873·64	7893·31	7913·00	7932·74	7952·47	7972·25	7992·03	100
101	8011·86	8031·69	8051·58	8071·46	8091·39	8111·32	8131·30	8151·28	101
102	8171·30	8191·33	8211·41	8231·49	8251·61	8271·73	8291·91	8312·09	102
103	8332·31	8352·53	8372·81	8393·08	8413·40	8433·73	8454·09	8474·47	103
104	8494·89	8515·31	8535·78	8556·25	8576·76	8597·28	8617·85	8638·42	104
105	8659·03	...	8700·32	...	8741·70	...	8783·18	...	105
106	8824·75	...	8866·43	...	8908·20	...	8950·07	...	106
107	8992·04	...	9034·11	...	9076·28	...	9118·54	...	107
108	9160·91	...	9203·37	...	9245·93	...	9288·58	...	108
109	9331·34	...	9374·19	...	9417·14	...	9460·19	...	109
110	9503·34	...	9546·59	...	9589·93	...	9633·37	...	110

Table CXCII.—Areas of Circles—*continued.*

Diameters.	Areas.								Diameters.
	·0	·⅛	·¼	·⅜	·½	·⅝	·¾	·⅞	
111	9676·91	...	9720·55	...	9764·29	...	9808·12	...	111
112	9852·06	...	9896·09	...	9940·22	...	9984·45	...	112
113	10028·77	...	10073·20	...	10117·72	...	10162·34	...	113
114	10207·06	...	10251·88	...	10296·79	...	10341·80	...	114
115	10386·91	...	10432·12	...	10477·43	...	10522·84	...	115
116	10568·34	...	10613·94	...	10659·65	...	10705·44	...	116
117	10751·34	...	10797·34	...	10843·43	...	10889·62	...	117
118	10935·91	...	10982·30	...	11028·78	...	11075·37	...	118
119	11122·05	...	11168·83	...	11215·71	...	11262·69	...	119
120	11309·76	...	11356·93	...	11404·20	...	11451·57	...	120
121	11499·04	...	11546·61	...	11594·27	...	11642·03	...	121
122	11689·89	...	11737·85	...	11785·91	...	11834·06	...	122
123	11882·32	...	11930·67	...	11979·12	...	12027·66	...	123
124	12076·31	...	12125·05	...	12173·90	...	12222·84	...	124
125	12271·87	...	...	...	12370·25	...	...	...	125
126	12469·01	...	...	...	12568·17	...	...	...	126
127	12667·72	...	...	...	12767·66	...	...	...	127
128	12867·99	...	...	...	12968·72	...	...	...	128
129	13069·84	...	...	...	13171·35	...	...	...	129
130	13273·26	...	...	...	13375·56	...	...	...	130
131	13478·25	...	...	...	13581·33	...	...	...	131
132	13684·81	...	...	...	13788·68	...	...	...	132

Table CXCII.—Areas of Circles—*continued.*

Diameters.	Areas.								Diameters.
	·0	·⅛	·¼	·⅜	·½	·⅝	·¾	·⅞	
89	6221·15	6238	...	...	13997·60	...	...	...	133
90	6361·74	637[illegible]	...	...	14208·08	...	...	...	134
91	6503·90	[illegible]	...	...	14420·14	...	...	...	135
92	6647·62	...	...	...	14633·77	...	...	...	136
93	6792·17	...	...	...	14848·97	...	...	...	137
94	14957·16	...	...	...	15065·74	...	...	...	138
139	15174·71	...	...	...	15284·08	...	...	...	139
140	15393·84	...	...	...	15503·99	...	...	...	140
141	15614·54	...	...	...	15725·48	...	...	...	141
142	15836·81	...	...	...	15948·53	...	...	...	142
143	16060·64	...	...	...	16173·15	...	...	...	143
144	16286·05	...	...	...	16399·35	...	...	...	144
145	16513·03	...	...	...	16627·11	...	...	...	145
146	16741·59	...	...	...	16856·45	...	...	...	146
147	16971·71	...	...	...	17087·36	...	...	...	147
148	17203·40	...	...	...	17319·84	...	...	...	148
149	17436·67	...	...	...	17553·89	...	...	...	149
150	17671·50	...	...	...	17789·51	...	...	...	150

Where the diameter contains an odd sixteenth, take one fourth of the area of twice the diameter; *e.g.*, area of $2\frac{1}{16} = \frac{\text{Area of } 4\frac{1}{8}}{4}$, &c. Areas of intermediate sizes may also be obtained by shifting the decimal point; *e.g.*, area, 132 = 13684·81; area of 13·2 = 136·8481; area of 1·32 = 1·368481, &c.

Table CXCIII.—Circumferences of Circles.

Diameters	Circumferences.								Diameters
	·0	·⅛	·¼	·⅜	·½	·⅝	·¾	·⅞	
0	·0	·3927	·7854	1·1781	1·5708	1·9635	2·3562	2·7489	0
1	3·1416	3·5343	3·9270	4·3197	4·7124	5·1051	5·4978	5·8905	1
2	6·2832	6·6759	7·0686	7·4613	7·8540	8·2467	8·6394	9·0321	2
3	9·4248	9·8175	10·2102	10·6029	10·9956	11·3883	11·7810	12·1737	3
4	12·5664	12·9591	13·3518	13·7445	14·1372	14·5299	14·9226	15·3153	4
5	15·7080	16·1007	16·4934	16·8861	17·2788	17·6715	18·0642	18·4569	5
6	18·8496	19·2423	19·6350	20·0277	20·4204	20·8131	21·2058	21·5985	6
7	21·9912	22·3839	22·7766	23·1693	23·5620	23·9547	24·3474	24·7401	7
8	25·1328	25·5255	25·9182	26·3109	26·7036	27·0963	27·4890	27·8817	8
9	28·2744	28·6671	29·0598	29·4525	29·8452	30·2379	30·6306	31·0233	9
10	31·4160	31·8087	32·2014	32·5941	32·9868	33·3795	33·7722	34·1649	10
11	34·5576	34·9503	35·3430	35·7357	36·1284	36·5211	36·9138	37·3065	11
12	37·6992	38·0919	38·4846	38·8773	39·2700	39·6627	40·0554	40·4481	12
13	40·8408	41·2335	41·6262	42·0189	42·4116	42·8043	43·1970	43·5897	13
14	43·9824	44·3751	44·7678	45·1605	45·5532	45·9459	46·3386	46·7313	14
15	47·1240	47·5167	47·9094	48·3021	48·6948	49·0875	49·4802	49·8729	15
16	50·2656	50·6583	51·0510	51·4437	51·8364	52·2291	52·6218	53·0145	16
17	53·4072	53·7999	54·1926	54·5853	54·9780	55·3707	55·7634	56·1561	17
18	56·5488	56·9415	57·3342	57·7269	58·1196	58·5123	58·9050	59·2977	18
19	59·6904	60·0831	60·4758	60·8685	61·2612	61·6539	62·0466	62·4393	19
20	62·8320	63·2247	63·6174	64·0101	64·4028	64·7955	65·1882	65·5809	20
21	65·9736	66·3663	66·7590	67·1517	67·5444	67·9371	68·3298	68·7225	21
22	69·1152	69·5079	69·9006	70·2933	70·6860	71·0787	71·4714	71·8641	22

Table CXCIII.—Circumferences of Circles—*continued.*

Diameters.	Circumferences.								Diameters.
	·0	·⅛	·¼	·⅜	·½	·⅝	·¾	·⅞	
23	72·2568	72·6495	73·0422	73·4349	73·8276	74·2203	74·6130	75·0057	23
24	75·3984	75·7911	76·1838	76·5765	76·9692	77·3619	77·7546	78·1473	24
25	78·5400	78·9327	79·3254	79·7181	80·1108	80·5035	80·8962	81·2889	25
26	81·6816	82·0743	82·4670	82·8597	83·2524	83·6451	84·0378	84·4305	26
27	84·8232	85·2159	85·6086	86·0013	86·3940	86·7867	87·1794	87·5721	27
28	87·9648	88·3575	88·7502	89·1429	89·5356	89·9283	90·3210	90·7137	28
29	91·1064	91·4991	91·8918	92·2845	92·6772	93·0699	93·4626	93·8553	29
30	94·2480	94·6407	95·0334	95·4261	95·8188	96·2115	96·6042	96·9969	30
31	97·3896	97·7823	98·1750	98·5677	98·9604	99·3531	99·7458	100·1385	31
32	100·5312	100·9239	101·3166	101·7093	102·1020	102·4947	102·8874	103·2801	32
33	103·6728	104·0655	104·4582	104·8509	105·2436	105·6363	106·0290	106·4217	33
34	106·814	107·207	107·600	107·992	108·385	108·778	109·171	109·563	34
35	109·956	110·349	110·741	111·134	111·527	111·919	112·312	112·705	35
36	113·098	113·490	113·883	114·276	114·668	115·061	115·454	115·846	36
37	116·239	116·632	117·025	117·417	117·810	118·203	118·595	118·988	37
38	119·381	119·773	120·166	120·559	120·952	121·344	121·737	122·130	38
39	122·522	122·915	123·308	123·700	124·093	124·486	124·879	125·271	39
40	125·664	126·057	126·449	126·842	127·235	127·627	128·020	128·413	40
41	128·806	129·198	129·591	129·984	130·376	130·769	131·162	131·554	41
42	131·947	132·340	132·733	133·125	133·518	133·911	134·303	134·696	42
43	135·089	135·481	135·874	136·267	136·660	137·052	137·445	137·838	43
44	138·230	138·623	139·016	139·408	139·801	140·194	140·587	140·979	44

Table CXCIII.—Circumferences of Circles—*continued.*

Diameters.	Circumferences.								Diameters.
	·0	·⅛	·¼	·⅜	·½	·⅝	·¾	·⅞	
45	141·372	141·765	142·157	142·550	142·943	143·335	143·728	144·121	45
46	144·514	144·906	145·299	145·692	146·084	146·477	146·870	147·262	46
47	147·655	148·048	148·441	148·833	149·226	149·619	150·011	150·404	47
48	150·797	151·189	151·582	151·975	152·368	152·760	153·153	153·546	48
49	153·938	154·331	154·724	155·116	155·509	155·902	156·295	156·687	49
50	157·080	157·473	157·865	158·258	158·651	159·043	159·436	159·829	50
51	160·222	160·614	161·207	161·400	161·792	162·185	162·578	162·970	51
52	163·363	163·756	164·149	164·541	164·934	165·327	165·719	166·112	52
53	166·505	166·897	167·290	167·683	168·076	168·468	168·861	169·254	53
54	169·646	170·039	170·432	170·824	171·217	171·610	172·003	172·395	54
55	172·788	173·181	173·573	173·966	174·359	174·751	175·144	175·537	55
56	175·930	176·322	176·715	177·108	177·500	177·893	178·286	178·678	56
57	179·071	179·464	179·857	180·249	180·642	181·035	181·427	181·820	57
58	182·213	182·605	182·998	183·391	183·784	184·176	184·569	184·962	58
59	185·354	185·747	186·140	186·532	186·925	187·318	187·711	188·103	59
60	188·496	188·889	189·281	189·674	190·067	190·459	190·852	191·245	60
61	191·638	192·030	192·423	192·816	193·208	193·601	193·994	194·396	61
62	194·779	195·172	195·565	195·957	196·350	196·743	197·135	197·528	62
63	197·921	198·313	198·706	199·099	199·492	199·884	200·277	200·670	63
64	201·062	201·455	201·848	202·240	202·633	203·026	203·419	203·811	64
65	204·204	204·597	204·989	205·382	205·775	206·167	206·560	206·953	65
66	207·346	207·738	208·131	208·524	208·916	209·309	209·702	210·094	66

Table CXCIII.—Circumferences of Circles—*continued.*

Diameters.	Circumferences.								Diameters.
	·0	·⅛	·¼	·⅜	·½	·⅝	·¾	·⅞	
67	210·487	210·880	211·273	211·665	212·058	212·451	212·843	213·236	67
68	213·629	214·021	214·414	214·807	215·200	215·592	215·985	216·378	68
69	216·770	217·163	217·556	217·948	218·341	218·734	219·127	219·519	69
70	219·912	220·305	220·697	221·090	221·483	221·875	222·268	222·661	70
71	223·054	223·446	223·839	224·232	224·624	225·017	225·410	225·802	71
72	226·195	226·588	226·981	227·373	227·766	228·158	228·551	228·944	72
73	229·337	229·729	230·122	230·515	230·908	231·300	231·693	232·086	73
74	232·478	232·871	233·264	233·656	234·049	234·442	234·835	235·227	74
75	235·620	236·013	236·405	236·798	237·191	237·583	237·976	238·369	75
76	238·762	239·154	239·547	239·940	240·332	240·725	241·118	241·510	76
77	241·903	242·296	242·689	243·081	243·474	243·867	244·259	244·652	77
78	245·045	245·437	245·830	246·223	246·616	247·008	247·401	247·794	78
79	248·186	248·579	248·972	249·364	249·757	250·150	250·543	250·935	79
80	251·328	251·721	252·113	252·506	252·899	253·291	253·684	254·077	80
81	254·470	254·862	255·255	255·648	256·040	256·433	256·826	257·218	81
82	257·611	258·004	258·397	258·789	259·182	259·575	259·967	260·360	82
83	260·753	261·145	261·538	261·931	262·324	262·716	263·109	263·502	83
84	263·894	264·287	264·680	265·072	265·465	265·858	266·251	266·643	84
85	267·036	267·429	267·821	268·214	268·607	268·999	269·392	269·785	85
86	270·178	270·570	270·963	271·356	271·748	272·141	272·534	272·926	86
87	273·319	273·712	274·105	274·497	274·890	275·283	275·675	276·068	87
88	276·461	276·853	277·246	277·629	278·032	278·424	278·817	279·210	88

Table CXCIII.—Circumferences of Circles—*continued.*

Diameters.	Circumferences.								Diameters.
	·0	·⅛	·¼	·⅜	·½	·⅝	·¾	·⅞	
89	279·602	279·995	280·388	280·780	281·173	281·566	281·959	282·351	89
90	282·744	283·137	283·529	283·922	284·315	284·707	285·100	285·493	90
91	285·886	286·278	286·671	287·064	287·456	287·849	288·242	288·634	91
92	289·027	289·420	289·813	290·205	290·598	290·991	291·383	291·776	92
93	292·169	292·562	292·954	293·347	293·740	294·132	294·525	294·918	93
94	295·310	295·703	296·096	296·488	296·881	297·274	297·667	298·059	94
95	298·452	298·845	299·237	299·630	300·023	300·415	300·808	301·201	95
96	301·594	301·986	302·379	302·772	303·164	303·557	303·950	304·342	96
97	304·735	305·128	305·521	305·913	306·306	306·699	307·091	307·484	97
98	307·877	308·270	308·662	309·055	309·448	309·840	310·233	310·626	98
99	311·018	311·411	311·804	312·196	312·589	312·982	313·375	313·767	99
100	314·160	314·553	314·945	315·338	315·731	316·123	316·516	316·909	100
101	317·302	317·694	318·087	318·480	318·872	319·265	319·658	320·050	101
102	320·443	320·836	321·229	321·621	322·014	322·407	322·799	323·192	102
103	323·585	323·977	324·370	324·763	325·156	325·548	325·941	326·334	103
104	326·726	327·119	327·512	327·904	328·297	328·690	329·083	329·475	104
105	329·868	...	330·653	...	331·439	...	332·224	...	105
106	333·010	...	333·795	...	334·580	...	335·366	...	106
107	336·151	...	336·937	...	337·722	...	338·507	...	107
108	339·293	...	340·078	...	340·864	...	341·649	...	108
109	342·434	...	343·220	...	344·005	...	344·791	...	109
110	345·576	...	346·361	...	347·147	...	347·932	...	110

Table CXCIII.—Circumferences of Circles—*continued.*

Diameters.	Circumferences.								Diameters.
	·0	·⅛	·¼	·⅜	·½	·⅝	·¾	·⅞	
111	348·718	...	349·503	...	350·288	...	351·074	...	111
112	351·859	...	352·645	...	353·430	...	354·215	...	112
113	355·001	...	355·786	...	356·572	...	357·357	...	113
114	358·142	...	358·928	...	359·713	...	360·499	...	114
115	361·284	...	362·069	...	362·855	...	363·640	...	115
116	364·426	...	365·211	...	365·996	...	366·782	...	116
117	367·567	...	368·353	...	369·138	...	369·923	...	117
118	370·709	...	371·494	...	372·280	...	373·065	...	118
119	373·850	...	374·636	...	375·421	...	376·207	...	119
120	376·992	...	377·777	...	378·563	...	379·348	...	120
121	380·134	...	380·919	...	381·704	...	382·490	...	121
122	383·275	...	384·061	...	384·846	...	385·631	...	122
123	386·417	...	387·202	...	387·988	...	388·773	...	123
124	389·558	...	390·344	...	391·129	...	391·915	...	124
125	392·700	...	...	...	394·271	...	...	...	125
126	395·842	...	...	...	397·412	...	...	...	126
127	398·983	...	...	...	400·554	...	...	...	127
128	402·125	...	...	...	403·696	...	...	...	128
129	405·266	...	...	...	406·837	...	...	...	129
130	408·408	...	...	...	409·979	...	...	...	130
131	411·550	...	...	...	413·120	...	...	...	131
132	414·691	...	...	...	416·262	...	...	...	132

Table CXCIII.—Circumferences of Circles—*continued.*

Diameters.	Circumferences.								Diameters.
	·0	·⅛	·¼	·⅜	·½	·⅝	·¾	·⅞	
133	417·833	...	...	...	419·404	...	...	...	133
134	420·974	...	...	...	422·545	...	...	...	134
135	424·116	...	...	...	425·687	...	...	...	135
136	427·258	...	...	...	428·828	...	...	...	136
137	430·399	...	...	...	431·970	...	...	...	137
138	433·541	...	...	...	435·112	...	...	...	138
139	436·682	...	...	...	438·253	...	...	...	139
140	439·824	...	...	...	441·395	...	...	...	140
141	442·966	...	...	...	444·536	...	...	...	141
142	446·107	...	...	...	447·678	...	...	...	142
143	449·249	...	...	...	450·820	...	...	...	143
144	452·390	...	...	...	453·961	...	...	...	144
145	455·532	...	...	...	457·103	...	...	...	145
146	458·674	...	...	...	460·244	...	...	...	146
147	461·815	...	...	...	463·386	...	...	...	147
148	464·957	...	...	...	466·528	...	...	...	148
149	468·098	...	...	...	469·669	...	...	...	149
150	471·240	...	...	...	472·811	...	...	...	150

Table CXCIV.—Areas and Circumferences of Small Circles.

Diameter.	Area.	Circum.	Diameter.	Area.	Circum.
1/32	·000767	·09817	1 1/32	·83525	3·2398
1/16	·003068	·19635	1 1/16	·88665	3·3379
3/32	·006903	·29452	1 3/32	·93956	3·4361
1/8	·012272	·39270	1 1/8	·99402	3·5343
5/32	·019175	·49087	1 5/32	1·0500	3·6325
3/16	·027612	·58905	1 3/16	1·1075	3·7306
7/32	·037583	·68722	1 7/32	1·1666	3·8288
1/4	·049087	·78540	1 1/4	1·2272	3·9270
9/32	·062126	·88357	1 9/32	1·2893	4·0252
5/16	·076699	·98175	1 5/16	1·3530	4·1233
11/32	·092806	1·0799	1 11/32	1·4182	4·2215
3/8	·11045	1·1781	1 3/8	1·4849	4·3197
13/32	·12962	1·2763	1 13/32	1·5531	4·4179
7/16	·15033	1·3744	1 7/16	1·6229	4·5160
15/32	·17257	1·4726	1 15/32	1·6943	4·6142
1/2	·19635	1·5708	1 1/2	1·7671	4·7124
17/32	·22166	1·6690	1 17/32	1·8415	4·8106
9/16	·24850	1·7671	1 9/16	1·9175	4·9087
19/32	·27688	1·8653	1 19/32	1·9949	5·0069
5/8	·30680	1·9635	1 5/8	2·0739	5·1051
21/32	·33824	2·0617	1 21/32	2·1545	5·2033
11/16	·37122	2·1598	1 11/16	2·2365	5·3014
23/32	·40574	2·2580	1 23/32	2·3201	5·3996
3/4	·44179	2·3562	1 3/4	2·4053	5·4978
25/32	·47937	2·4544	1 25/32	2·4919	5·5959
13/16	·51849	2·5525	1 13/16	2·5801	5·6941
27/32	·55914	2·6507	1 27/32	2·6699	5·7923
7/8	·60132	2·7489	1 7/8	2·7612	5·8905
29/32	·64504	2·8471	1 29/32	2·8540	5·9886
15/16	·69029	2·9452	1 15/16	2·9483	6·0868
31/32	·73708	3·0434	1 31/32	3·0442	6·1850
1	·78540	3·1416	2	3·1416	6·2832

Table CXCV.—Spheres, Volume and Surface of.

(Volume $=\frac{\pi d^3}{6}=\frac{d^3}{1{\cdot}91}$; surface $=\pi d^2$.)

Dia.	Surface.	Volume.	Dia.	Surface.	Volume.	Dia.	Surface.	Volume.
1	3·142	0·524	41	5281·0	36,084	81	20,612	278,241
2	12·546	4·189	42	5541·8	38,785	82	21,120	288,640
3	28·274	14·136	43	5808·8	41,627	83	21,633	299,361
4	50·266	33·508	44	6082·1	44,597	84	22,146	310,316
5	78·540	65·445	45	6361·7	47,707	85	22,704	321,531
6	113·098	113·09	46	6647·6	50,961	86	23,227	333,013
7	153·938	179·58	47	6939·8	54,356	87	23,753	344,764
8	201·062	268·06	48	7238·3	57,900	88	24,328	356,791
9	254·47	381·67	49	7543·0	61,596	89	24,860	369,089
10	314·16	523·56	50	7854·0	65,445	90	25,434	381,675
11	380·13	696·86	51	8171·3	69,450	91	26,016	394,540
12	452·39	904·71	52	8494·9	73,612	92	26,601	407,690
13	530·93	1150·3	53	8824·8	77,946	93	27,150	421,130
14	615·75	1436·7	54	9160·9	82,442	94	27,742	434,860
15	706·86	1767·0	55	9503·3	87,105	95	28,338	448,887
16	804·25	2144·5	56	9852·1	91,945	96	28,939	463,212
17	907·92	2572·2	57	10,207	96,960	97	29,543	477,840
18	1017·9	3053·4	58	10,568	102,152	98	30,150	492,771
19	1134·1	3591·1	59	10,936	107,523	99	30,787	508,010
20	1256·6	4188·5	60	11,310	113,089	100	31,416	523,560
21	1385·5	4848	61	11,690	118,847	101	32,036	539,424
22	1520·5	5575	62	12,076	124,780	102	32,685	555,602
23	1661·9	6370	63	12 469	130,914	103	33,329	572,109
24	1809·6	7237	64	12,868	137,248	104	33,980	588,932
25	1963·5	8181	65	13,273	143,783	105	34,636	606,086
26	2123·7	9202	66	13,685	150,521	106	35,299	623,570
27	2290·2	10,305	67	14,103	157,466	107	35,968	641,384
28	2463·0	11,490	68	14,527	164,623	108	36,644	659,535
29	2642·1	12,769	69	14,957	172,000	109	37,325	678,025
30	2827·4	14,136	70	15,394	179,581	110	38,013	696,860
31	3019·1	15,597	71	15,837	187,387	111	38,708	716,037
32	3217·0	17,152	72	16,286	195,418	112	39,408	735,560
33	3421·2	18,812	73	16,742	203,674	113	40,116	755,440
34	3631·7	20,523	74	17,204	212,193	114	40,828	775,676
35	3848·5	22,377	75	17,672	220,878	115	41,548	796,270
36	4071·5	24,372	76	18,146	229,827	116	42,273	817,223
37	4300·9	26,466	77	18,826	239,023	117	43,005	838,535
38	4536·5	28,675	78	19,114	248,456	118	43,744	860,227
39	4778·4	31,057	79	19,607	258,135	119	44,488	882,282
40	5026·6	33,508	80	20,106	268,063	120	45,240	904,712

Table CXCV.—Spheres, Volume and Surface of—*continued.*

$$\left(\text{Volume} = \frac{\pi d^3}{6} = \frac{d^3}{1{\cdot}91}\ ;\ \text{surface} = \pi d^2.\right)$$

Dia.	Surface.	Volume.	Dia.	Surface.	Volume.	Dia.	Surface.	Volume.
121	45,996	927,519	131	53,913	1,177,000	141	62,460	1,467,644
122	46,760	940,413	132	54,740	1,204,172	142	63,348	1,500,000
123	47,529	974,276	133	55,572	1,231,750	143	64,244	1,531,100
124	48,305	998,230	134	56,412	1,260,000	144	65,144	1,563,342
125	49,088	1,022,577	135	57,256	1,288,152	145	66,052	1,596,131
126	49,876	1,047,320	136	58,108	1,317,000	146	66,968	1,629,390
127	50,672	1,072,452	137	58,965	1,346,251	147	67,888	1,663,100
128	51,472	1,098,000	138	59,829	1,376,000	148	68,813	1,700,000
129	52,280	1,123,916	139	60,700	1,406,083	149	69,748	1,731,900
130	53,092	1,150,262	140	61,576	1,436,650	150	70,688	1,767,000

Note.—Cones have half the volume and 0·809 of the surface of corresponding spheres, that is, cones having a diameter of base and height each $= d$, volume $= d^3 \div 3{\cdot}82$, surface $= d^2 \times 2{\cdot}5416$.

STOWAGE.

Miscellaneous Weights and Measures.

Coal.

1 sack = 224 lbs.
1 chald. = 12 sacks.
1 Newcastle chald. = 23½ sacks.
1 keel = 8 Newcastle chald.

Welsh	42·7	cubic ft. to ton.	Lancashire	45·2	cubic ft. to ton.
Newcastle	45·3	,, ,,	Scotch	42	,, ,,
S. Yorkshire	46	,, ,,	American	42·4	,, ,,

Grain.

480 lbs. = 1 quarter (approximate).

Oats	56	bushels	= 1 ton	,,
,,	1	,,	= 40 lbs.	,,
Wheat	37⅓	,,	= 1 ton	,,
,,	1	,,	= 60 lbs.	,,
Maize	40	,,	= 1 ton	,,

Cotton (American).

4½ bales = 1 ton (measurement, approximate).
2⅜ to 2¼ bales = 1 ton oil-cake or meal ,,
1 bale = 20 bushels wheat ,,

Table CXCVI.—Squares, Cubes, Square Roots, Cube Roots, and Reciprocals of all Integer Numbers from 1 to 2200.

No.	Square	Cube	Square Root	Cube Root	Reciprocal
1	1	1	1·0000000	1·0000000	1·000000000
2	4	8	1·4142136	1·2599210	·500000000
3	9	27	1·7320508	1·4422496	·333333333
4	16	64	2·0000000	1·5874011	·250000000
5	25	125	2·2360680	1·7099759	·200000000
6	36	216	2·4494897	1·8171206	·166666667
7	49	343	2·6457513	1·9129312	·142857143
8	64	512	2·8284271	2·0000000	·125000000
9	81	729	3·0000000	2·0800837	·111111111
10	100	1000	3·1622777	2·1544347	·100000000
11	121	1331	3·3166248	2·2239801	·090909091
12	144	1728	3·4641016	2·2894286	·083333333
13	169	2197	3·6055513	2·3513347	·076923077
14	196	2744	3·7416574	2·4101422	·071428571
15	225	3375	3·8729833	2·4662121	·066666667
16	256	4096	4·0000000	2·5198421	·062500000
17	289	4913	4·1231056	2·5712816	·058823529
18	324	5832	4·2426407	2·6207414	·055555556
19	361	6859	4·3588989	2·6684016	·052631579
20	400	8000	4·4721360	2·7144177	·050000000
21	441	9261	4·5825757	2·7589243	·047619048
22	484	10648	4·6904158	2·8020393	·045454545
23	529	12167	4·7958315	2·8438670	·043478261
24	576	13824	4·8989795	2·8844991	·041666667
25	625	15625	5·0000000	2·9240177	·040000000
26	676	17576	5·0990195	2·9624960	·038461538
27	729	19683	5·1961524	3·0000000	·037037037
28	784	21952	5·2915026	3·0365889	·035714286
29	841	24389	5·3851648	3·0723168	·034482759
30	900	27000	5·4772256	3·1072325	·033333333
31	961	29791	5·5677644	3·1413806	·032258065
32	1024	32768	5·6568542	3·1748021	·031250000
33	1089	35937	5·7445626	3·2075343	·030303030
34	1156	39304	5·8309519	3·2396118	·029411765
35	1225	42875	5·9160798	3·2710663	·028571429
36	1296	46656	6·0000000	3·3019272	·027777778
37	1369	50653	6·0827625	3·3322218	·027027027
38	1444	54872	6·1644140	3·3619754	·026315789
39	1521	59319	6·2449980	3·3912114	·025641026
40	1600	64000	6·3245553	3·4199519	·025000000
41	1681	68921	6·4031242	3·4482172	·024390244
42	1764	74088	6·4807407	3·4760266	·023809524
43	1849	79507	6·5574385	3·5033981	·023255814
44	1936	85184	6·6332496	3·5303483	·022727273
45	2025	91125	6·7082039	3·5568933	·022222222

No.	Square	Cube	Square Root	Cube Root	Reciprocal
46	2116	97336	6·7823300	3·5830479	·021739130
47	2209	103823	6·8556546	3·6088261	·021276600
48	2304	110592	6·9282032	3·6342411	·020833333
49	2401	117649	7·0000000	3·6593057	·020408163
50	2500	125000	7·0710678	3·6840314	·020000000
51	2601	132651	7·1414284	3·7084298	·019607843
52	2704	140608	7·2111026	3·7325111	·019230769
53	2809	148877	7·2801099	3·7562858	·018867925
54	2916	157464	7·3484692	3·7797631	·018518519
55	3025	166375	7·4161985	3·8029525	·018181818
56	3136	175616	7·4833148	3·8258624	·017857143
57	3249	185193	7·5498344	3·8485011	·017543860
58	3364	195112	7·6157731	3·8708766	·017241379
59	3481	205379	7·6811457	3·8929965	·016949153
60	3600	216000	7·7459667	3·9148676	·016666667
61	3721	226981	7·8102497	3·9364972	·016393443
62	3844	238328	7·8740079	3·9578915	·016129032
63	3969	250047	7·9372539	3·9790571	·015873016
64	4096	262144	8·0000000	4·0000000	·015625000
65	4225	274625	8·0622577	4·0207256	·015384615
66	4356	287496	8·1240384	4·0412401	·015151515
67	4489	300763	8·1853528	4·0615480	·014925373
68	4624	314432	8·2462113	4·0816551	·014705882
69	4761	328509	8·3066239	4·1015661	·014492754
70	4900	343000	8·3666003	4·1212853	·014285714
71	5041	357911	8·4261498	4·1408178	·014084507
72	5184	373248	8·4852814	4·1601676	·013888889
73	5329	389017	8·5440037	4·1793392	·013698630
74	5476	405224	8·6023253	4·1983364	·013513514
75	5625	421875	8·6602540	4·2171633	·013333333
76	5776	438976	8·7177979	4·2358236	·013157895
77	5929	456533	8·7749644	4·2543210	·012987013
78	6084	474552	8·8317609	4·2726586	·012820513
79	6241	493039	8·8881944	4·2908404	·012658228
80	6400	512000	8·9442719	4·3088695	·012500000
81	6561	531441	9·0000000	4·3267487	·012345679
82	6724	551368	9·0553851	4·3444815	·012195122
83	6889	571787	9·1104336	4·3620707	·012048193
84	7056	592704	9·1651514	4·3795191	·011904762
85	7225	614125	9·2195445	4·3968296	·011764706
86	7396	636056	9·2736185	4·4140049	·011627907
87	7569	658503	9·3273791	4·4310476	·011494253
88	7744	681472	9·3808315	4·4479602	·011363636
89	7921	704969	9·4339811	4·4647451	·011235955
90	8100	729000	9·4868330	4·4814047	·011111111
91	8281	753571	9·5393920	4·4979414	·010989011
92	8464	778688	9·5916630	4·5143574	·010869565
93	8649	804357	9·6436508	4·5306549	·010752688
94	8836	830584	9·6953597	4·5468359	·010638298

No.	Square	Cube	Square Root	Cube Root	Reciprocal
95	9025	857375	9·7467943	4·5629026	·010526316
96	9216	884736	9·7979590	4·5788570	·010416667
97	9409	912673	9·8488578	4·5947009	·010309278
98	9604	941192	9·8994949	4·6104363	·010204082
99	9801	970299	9·9498744	4·6260650	·010101010
100	10000	1000000	10·0000000	4·6415888	·010000000
101	10201	1030301	10·0498756	4·6570095	·009900990
102	10404	1061208	10·0995049	4·6723287	·009803922
103	10609	1092727	10·1488916	4·6875482	·009708738
104	10816	1124864	10·1980390	4·7026694	·009615385
105	11025	1157625	10·2469508	4·7176940	·009523810
106	11236	1191016	10·2956301	4·7326235	·009433962
107	11449	1225043	10·3440804	4·7474594	·009345794
108	11664	1259712	10·3923048	4·7622032	·009259259
109	11881	1295029	10·4403065	4·7768562	·009174312
110	12100	1331000	10·4880885	4·7914199	·009090909
111	12321	1367631	10·5356538	4·8058955	·009009009
112	12544	1404928	10·5830052	4·8202845	·008928571
113	12769	1442897	10·6301458	4·8345881	·008849558
114	12996	1481544	10·6770783	4·8488076	·008771930
115	13225	1520875	10·7238053	4·8629442	·008695652
116	13456	1560896	10·7703296	4·8769990	·008620690
117	13689	1601613	10·8166538	4·8909732	·008547009
118	13924	1643032	10·8627805	4·9048681	·008474576
119	14161	1685159	10·9087121	4·9186847	·008403361
120	14400	1728000	10·9544512	4·9324242	·008333333
121	14641	1771561	11·0000000	4·9460874	·008264463
122	14884	1815848	11·0453610	4·9596757	·008196721
123	15129	1860867	11·0905365	4·9731898	·008130081
124	15376	1906624	11·1355287	4·9866310	·008064516
125	15625	1953125	11·1803399	5·0000000	·008000000
126	15876	2000376	11·2249722	5·0132979	·007936508
127	16129	2048383	11·2694277	5·0265257	·007874016
128	16384	2097152	11·3137085	5·0396842	·007812500
129	16641	2146689	11·3578167	5·0527743	·007751938
130	16900	2197000	11·4017543	5·0657970	·007692308
131	17161	2248091	11·4455231	5·0787531	·007633588
132	17424	2299968	11·4891253	5·0916434	·007575758
133	17689	2352637	11·5325626	5·1044687	·007518797
134	17956	2406104	11·5758369	5·1172299	·007462687
135	18225	2460375	11·6189500	5·1299278	·007407407
136	18496	2515456	11·6619038	5·1425632	·007352941
137	18769	2571353	11·7046999	5·1551367	·007299270
138	19044	2628072	11·7473401	5·1676493	·007246377
139	19321	2685619	11·7898261	5·1801015	·007194245
140	19600	2744000	11·8321596	5·1924941	·007142857
141	19881	2803221	11·8743422	5·2048279	·007092199
142	20164	2863288	11·9163753	5·2171034	·007042254
143	20449	2924207	11·9582607	5·2293215	·006993007

No.	Square	Cube	Square Root	Cube Root	Reciprocal
144	20736	2985984	12·0000000	5·2414828	·006944444
145	21025	3048625	12·0415946	5·2535879	·006896552
146	21316	3112136	12·0830460	5·2656374	·006849315
147	21609	3176523	12·1243557	5·2776321	·006802721
148	21904	3241792	12·1655251	5·2895725	·006756757
149	22201	3307949	12·2065556	5·3014592	·006711409
150	22500	3375000	12·2474487	5·3132928	·006666667
151	22801	3442951	12·2882057	5·3250740	·006622517
152	23104	3511808	12·3288280	5·3368033	·006578947
153	23409	3581577	12·3693169	5·3484812	·006535948
154	23716	3652264	12·4096736	5·3601084	·006493506
155	24025	3723875	12·4498996	5·3716854	·006451613
156	24336	3796416	12·4899960	5·3832126	·006410256
157	24649	3869893	12·5299641	5·3946907	·006369427
158	24964	3944312	12·5698051	5·4061202	·006329114
159	25281	4019679	12·6095202	5·4175015	·006289308
160	25600	4096000	12·6491106	5·4288352	·006250000
161	25921	4173281	12·6885775	5·4401218	·006211180
162	26244	4251528	12·7279221	5·4513618	·006172840
163	26569	4330747	12·7671453	5·4625556	·006134969
164	26896	4410944	12·8062485	5·4737037	·006097561
165	27225	4492125	12·8452326	5·4848066	·006060606
166	27556	4574296	12·8840987	5·4958647	·006024096
167	27889	4657463	12·9228480	5·5068784	·005988024
168	28224	4741632	12·9614814	5·5178484	·005952381
169	28561	4826809	13·0000000	5·5287748	·005917160
170	28900	4913000	13·0384048	5·5396583	·005882353
171	29241	5000211	13·0766968	5·5504991	·005847953
172	29584	5088448	13·1148770	5·5612978	·005813953
173	29929	5177717	13·1529464	5·5720546	·005780347
174	30276	5268024	13·1909060	5·5827702	·005747126
175	30625	5359375	13·2287566	5·5934447	·005714286
176	30976	5451776	13·2664992	5·6040787	·005681818
177	31329	5545233	13·3041347	5·6146724	·005649718
178	31684	5639752	13·3416641	5·6252263	·005617978
179	32041	5735339	13·3790882	5·6357408	·005586592
180	32400	5832000	13·4164079	5·6462162	·005555556
181	32761	5929741	13·4536240	5·6566528	·005524862
182	33124	6028568	13·4907376	5·6670511	·005494505
183	33489	6128487	13·5277493	5·6774114	·005464481
184	33856	6229504	13·5646600	5·6877340	·005434783
185	34225	6331625	13·6014705	5·6980192	·005405405
186	34596	6434856	13·6381817	5·7082675	·005376344
187	34969	6539203	13·6747943	5·7184791	·005347594
188	35344	6644672	13·7113092	5·7286543	·005319149
189	35721	6751269	13·7477271	5·7387936	·005291005
190	36100	6859000	13·7840488	5·7488971	·005263158
191	36481	6967871	13·8202750	5·7589652	·005235602
192	36864	7077888	13·8564065	5·7689982	·005208333

No.	Square	Cube	Square Root	Cube Root	Reciprocal
193	37249	7189057	13·8924440	5·7789966	·005181347
194	37636	7301384	13·9283883	5·7889604	·005154639
195	38025	7414875	13·9642400	5·7988900	·005128205
196	38416	7529536	14·0000000	5·8087857	·005102041
197	38809	7645373	14·0356688	5·8186479	·005076142
198	39204	7762392	14·0712473	5·8284767	·005050505
199	39601	7880599	14·1067360	5·8382725	·005025126
200	40000	8000000	14·1421356	5·8480355	·005000000
201	40401	8120601	14·1774469	5·8577660	·004975124
202	40804	8242408	14·2126704	5·8674643	·004950495
203	41209	8365427	14·2478068	5·8771307	·004926108
204	41616	8489664	14·2828569	5·8867653	·004901961
205	42025	8615125	14·3178211	5·8963685	·004878049
206	42436	8741816	14·3527001	5·9059406	·004854369
207	42849	8869743	14·3874946	5·9154817	·004830918
208	43264	8998912	14·4222051	5·9249921	·004807692
209	43681	9129329	14·4568323	5·9344721	·004784689
210	44100	9261000	14·4913767	5·9439220	·004761905
211	44521	9393931	14·5258390	5·9533418	·004739336
212	44944	9528128	14·5602198	5·9627320	·004716981
213	45369	9663597	14·5945195	5·9720926	·004694836
214	45796	9800344	14·6287388	5·9814240	·004672897
215	46225	9938375	14·6628783	5·9907264	·004651163
216	46656	10077696	14·6969385	6·0000000	·004629630
217	47089	10218313	14·7309199	6·0092450	·004608295
218	47524	10360232	14·7648231	6·0184617	·004587156
219	47961	10503459	14·7986486	6·0276502	·004566210
220	48400	10648000	14·8323970	6·0368107	·004545455
221	48841	10793861	14·8660687	6·0459435	·004524887
222	49284	10941048	14·8996644	6·0550489	·004504505
223	49729	11089567	14·9331845	6·0641270	·004484305
224	50176	11239424	14·9666295	6·0731779	·004464286
225	50625	11390625	15·0000000	6·0822020	·004444444
226	51076	11543176	15·0332964	6·0911994	·004424779
227	51529	11697083	15·0665192	6·1001702	·004405286
228	51984	11852352	15·0996689	6·1091147	·004385965
229	52441	12008989	15·1327460	6·1180332	·004366812
230	52900	12167000	15·1657509	6·1269257	·004347826
231	53361	12326391	15·1986842	6·1357924	·004329004
232	53824	12487168	15·2315462	6·1446337	·004310345
233	54289	12649337	15·2643375	6·1534495	·004291845
234	54756	12812904	15·2970585	6·1622401	·004273504
235	55225	12977875	15·3297097	6·1710058	·004255319
236	55696	13144256	15·3622915	6·1797466	·004237288
237	56169	13312053	15·3948043	6·1884628	·004219409
238	56644	13481272	15·4272486	6·1971544	·004201681
239	57121	13651919	15·4596248	6·2058218	·004184100
240	57600	13824000	15·4919334	6·2144650	·004166667
241	58081	13997521	15·5241747	6·2230843	·004149378

No.	Square	Cube	Square Root	Cube Root	Reciprocal
242	58564	14172488	15·5563492	6·2316797	·004132231
243	59049	14348907	15·5884573	6·2403515	·004115226
244	59536	14526784	15·6204994	6·2487998	·004098361
245	60025	14706125	15·6524758	6·2573248	·004081633
246	60516	14886936	15·6843871	6·2658266	·004065041
247	61009	15069223	15·7162336	6·2743054	·004048583
248	61504	15252992	15·7480157	6·2827613	·004032258
249	62001	15438249	15·7797338	6·2911946	·004016064
250	62500	15625000	15·8113883	6·2996053	·004000000
251	63001	15813251	15·8429795	6·3079935	·003984064
252	63504	16003008	15·8745079	6·3163596	·003968254
253	64009	16194277	15·9059737	6·3247035	·003952569
254	64516	16387064	15·9373775	6·3330256	·003937008
255	65025	16581375	15·9687194	6·3413257	·003921569
256	65536	16777216	16·0000000	6·3493042	·003906250
257	66049	16974593	16·0312195	6·3578611	·003891051
258	66564	17173512	16·0623784	6·3660968	·003875969
259	67081	17373979	16·0934769	6·3743111	·003861004
260	67600	17576000	16·1245155	6·3825043	·003846154
261	68121	17779581	16·1554944	6·3906765	·003831418
262	68644	17984728	16·1864141	6·3988279	·003816794
263	69169	18191447	16·2172747	6·4069585	·003802281
264	69696	18399744	16·2480768	6·4150687	·003787879
265	70225	18609625	16·2788206	6·4231583	·003773585
266	70756	18821096	16·3095064	6·4312276	·003759398
267	71289	19034163	16·3401346	6·4392767	·003745318
268	71824	19248832	16·3707055	6·4473057	·003731343
269	72361	19465109	16·4012195	6·4553148	·003717472
270	72900	19683000	16·4316767	6·4633041	·003703704
271	73441	19902511	16·4620776	6·4712736	·003690037
272	73984	20123648	16·4924225	6·4792236	·003676471
273	74529	20346417	16·5227116	6·4871541	·003663004
274	75076	20570824	16·5529454	6·4950653	·003649635
275	75625	20796875	16·5831240	6·5029572	·003636364
276	76176	21024576	16·6132477	6·5108300	·003623188
277	76729	21253933	16·6433170	6·5186839	·003610108
278	77284	21484952	16·6733320	6·5265189	·003597122
279	77841	21717639	16·7032931	6·5343351	·003584229
280	78400	21952000	16·7332005	6·5421326	·003571429
281	78961	22188041	16·7630546	6·5499116	·003558719
282	79524	22425768	16·7928556	6·5576722	·003546099
283	80089	22665187	16·8226038	6·5654144	·003533569
284	80656	22906304	16·8522995	6·5731385	·003521127
285	81225	23149125	16·8819430	6·5808443	·003508772
286	81796	23393656	16·9115345	6·5885323	·003496503
287	82369	23639903	16·9410743	6·5962023	·003484321
288	82944	23887872	16·9705627	6·6038545	·003472222
289	83521	24137569	17·0000000	6·6114890	·003460208
290	84100	24389000	17·0293864	6·6191060	·003448276

No.	Square	Cube	Square Root	Cube Root	Reciprocal
291	84681	24642171	17·0587221	6·6267054	·003436426
292	85264	24897088	17·0880075	6·6342874	·003424658
293	85849	25153757	17·1172428	6·6418522	·003412969
294	86436	25412184	17·1464282	6·6493998	·003401361
295	87025	25672375	17·1755640	6·6569302	·003389831
296	87616	25934336	17·2046505	6·6644437	·003378378
297	88209	26198073	17·2336879	6·6719403	·003367003
298	88804	26463592	17·2626765	6·6794200	·003355705
299	89401	26730899	17·2916165	6·6868831	·003344482
300	90000	27000000	17·3205081	6·6943295	·003333333
301	90601	27270901	17·3493516	6·7017593	·003322259
302	91204	27543608	17·3781472	6·7091729	·003311258
303	91809	27818127	17·4068952	6·7165700	·003300330
304	92416	28094464	17·4355958	6·7239508	·003289474
305	93025	28372625	17·4642492	6·7313155	·003278689
306	93636	28652616	17·4928557	6·7386641	·003267974
307	94249	28934443	17·5214155	6·7459967	·003257329
308	94864	29218112	17·5499288	6·7533134	·003246753
309	95481	29503629	17·5783958	6·7606143	·003236246
310	96100	29791000	17·6068169	6·7678995	·003225806
311	96721	30080231	17·6351921	6·7751690	·003215434
312	97344	30371328	17·6635217	6·7824229	·003205128
313	97969	30664297	17·6918060	6·7896613	·003194888
314	98596	30959144	17·7200451	6·7968844	·003184713
315	99225	31255875	17·7482393	6·8040921	·003174603
316	99856	31554496	17·7763888	6·8112847	·003164557
317	100489	31855013	17·8044938	6·8184620	·003154574
318	101124	32157432	17·8325545	6·8256242	·003144654
319	101761	32461759	17·8605711	6·8327714	·003134796
320	102400	32768000	17·8885438	6·8399037	·003125000
321	103041	33076161	17·9164729	6·8470213	·003115265
322	103684	33386248	17·9443584	6·8541240	·003105590
323	104329	33698267	17·9722008	6·8612120	·003095975
324	104976	34012224	18·0000000	6·8682855	·003086420
325	105625	34328125	18·0277564	6·8753443	·003076923
326	106276	34645976	18·0554701	6·8823888	·003067485
327	106929	34965783	18·0831413	6·8894188	·003058104
328	107584	35287552	18·1107703	6·8964345	·003048780
329	108241	35611289	18·1383571	6·9034359	·003039514
330	108900	35937000	18·1659021	6·9104232	·003030303
331	109561	36264691	18·1934054	6·9173964	·003021148
332	110224	36594368	18·2208672	6·9243556	·003012048
333	110889	36926037	18·2482876	6·9313008	·003003003
334	111556	37259704	18·2756669	6·9382321	·002994012
335	112225	37595375	18·3030052	6·9451496	·002985075
336	112896	37933056	18·3303028	6·9520533	·002976190
337	113569	38272753	18·3575598	6·9589434	·002967359
338	114244	38614472	18·3847763	6·9658198	·002958580
339	114921	38958219	18·4119526	6·9726826	·002949853

No.	Square	Cube	Square Root	Cube Root	Reciprocal
340	115600	39304000	18·4390889	6·9795321	·002941176
341	116281	39651821	18·4661853	6·9863681	·002932551
342	116964	40001688	18·4932420	6·9931906	·002923977
343	117649	40353607	18·5202592	7·0000000	·002915452
344	118336	40707584	18·5472370	7·0067962	·002906977
345	119025	41063625	18·5741756	7·0135791	·002898551
346	119716	41421736	18·6010752	7·0203490	·002890173
347	120409	41781923	18·6279360	7·0271058	·002881844
348	121104	42144192	18·6547581	7·0338497	·002873563
349	121801	42508549	18·6815147	7·0405806	·002865330
350	122500	42875000	18·7082869	7·0472987	·002857143
351	123201	43243551	18·7349940	7·0540041	·002849003
352	123904	43614208	18·7616630	7·0606967	·002840909
353	124609	43986977	18·7882942	7·0673767	·002832861
354	125316	44361864	18·8148877	7·0740440	·002824859
355	126025	44738875	18·8414437	7·0806988	·002816901
356	126736	45118016	18·8679623	7·0873411	·002808989
357	127449	45499293	18·8944436	7·0939709	·002801120
358	128164	45882712	18·9208879	7·1005885	·002793296
359	128881	46268279	18·9472953	7·1071937	·002785515
360	129600	46656000	18·9736660	7·1137866	·002777778
361	130321	47045881	19·0000000	7·1203674	·002770083
362	131044	47437928	19·0262976	7·1269360	·002762431
363	131769	47832147	19·0525589	7·1334925	·002754821
364	132496	48228544	19·0787840	7·1400370	·002747253
365	133225	48627125	19·1049732	7·1465695	·002739726
366	133956	49027896	19·1311265	7·1530901	·002732240
367	134689	49430863	19·1572441	7·1595988	·002724796
368	135424	49836032	19·1833261	7·1660957	·002717391
369	136161	50243409	19·2093727	7·1725809	·002710027
370	136900	50653000	19·2353841	7·1790544	·002702703
371	137641	51064811	19·2613603	7·1855162	·002695418
372	138384	51478848	19·2873015	7·1919663	·002688172
373	139129	51895117	19·3132079	7·1984050	·002680965
374	139876	52313624	19·3390796	7·2048322	·002673797
375	140625	52734375	19·3649167	7·2112479	·002666667
376	141376	53157376	19·3907194	7·2176522	·002659574
377	142129	53582633	19·4164878	7·2240450	·002652520
378	142884	54010152	19·4422221	7·2304268	·002645503
379	143641	54439939	19·4679223	7·2367972	·002638522
380	144400	54872000	19·4935887	7·2431565	·002631579
381	145161	55306341	19·5192213	7·2495045	·002624672
382	145924	55742968	19·5448203	7·2558415	·002617801
383	146689	56181887	19·5703858	7·2621675	·002610966
384	147456	56623104	19·5959179	7·2684824	·002604167
385	148225	57066625	19·6214169	7·2747864	·002597403
386	148996	57512456	19·6468827	7·2810794	·002590674
387	149769	57960603	19·6723156	7·2873617	·002583979
388	150544	58411072	19·6977156	7·2936330	·002577320

No.	Square	Cube	Square Root	Cube Root	Reciprocal
389	151321	58863869	19·7230829	7·2998936	·002570694
390	152100	59319000	19·7484177	7·3061436	·002564103
391	152881	59776471	19·7737199	7·3123828	·002557545
392	153664	60236288	19·7989899	7·3186114	·002551020
393	154449	60698457	19·8242276	7·3248295	·002544529
394	155236	61162984	19·8494332	7·3310369	·002538071
395	156025	61629875	19·8746069	7·3372339	·002531646
396	156816	62099136	19·8997487	7·3434205	·002525253
397	157609	62570773	19·9248588	7·3495966	·002518892
398	158404	63044792	19·9499373	7·3557624	·002512563
399	159201	63521199	19·9749844	7·3619178	·002506266
400	160000	64000000	20·0000000	7·3680630	·002500000
401	160801	64481201	20·0249844	7·3741979	·002493766
402	161604	64964808	20·0499377	7·3803227	·002487562
403	162409	65450827	20·0748599	7·3864373	·002481390
404	163216	65939264	20·0997512	7·3925418	·002475248
405	164025	66430125	20·1246118	7·3986363	·002469136
406	164836	66923416	20·1494417	7·4047206	·002463054
407	165649	67419143	20·1742410	7·4107950	·002457002
408	166464	67917312	20·1990099	7·4168595	·002450980
409	167281	68417929	20·2237484	7·4229142	·002444988
410	168100	68921000	20·2484567	7·4289589	·002439024
411	168921	69426531	20·2731349	7·4349938	·002433090
412	169744	69934528	20·2977831	7·4410189	·002427184
413	170569	70444997	20·3224014	7·4470342	·002421308
414	171396	70957944	20·3469899	7·4530399	·002415459
415	172225	71473375	20·3715488	7·4590359	·002409639
416	173056	71991296	20·3960781	7·4650223	·002403846
417	173889	72511713	20·4205779	7·4709991	·002398082
418	174724	73034632	20·4450483	7·4769664	·002392344
419	175561	73560059	20·4694895	7·4829242	·002386635
420	176400	74088000	20·4939015	7·4888724	·002380952
421	177241	74618461	20·5182845	7·4948113	·002375297
422	178084	75151448	20·5426386	7·5007406	·002369668
423	178929	75686967	20·5669638	7·5066607	·002364066
424	179776	76225024	20·5912603	7·5125715	·002358491
425	180625	76765625	20·6155281	7·5184730	·002352941
426	181476	77308776	20·6397674	7·5243652	·002347418
427	182329	77854483	20·6639783	7·5302482	·002341920
428	183184	78402752	20·6881609	7·5361221	·002336449
429	184041	78953589	20·7123152	7·5419867	·002331002
430	184900	79507000	20·7364414	7·5478423	·002325581
431	185761	80062991	20·7605395	7·5536888	·002320186
432	186624	80621568	20·7846097	7·5595263	·002314815
433	187489	81182737	20·8086520	7·5653548	·002309469
434	188356	81746504	20·8326667	7·5711743	·002304147
435	189225	82312875	20·8566536	7·5769849	·002298851
436	190096	82881856	20·8806130	7·5827865	·002293578
437	190969	83453453	20·9045450	7·5885793	·002288330

No.	Square	Cube	Square Root	Cube Root	Reciprocal
438	191844	84027672	20·9284495	7·5943633	·002283105
439	192721	84604519	20·9523268	7·6001385	·002277004
440	193600	85184000	20·9761770	7·6059049	·002272727
441	194481	85766121	21·0000000	7·6116626	·002267574
442	195364	86350888	21·0237960	7·6174116	·002262443
443	196249	86938307	21·0475652	7·6231519	·002257336
444	197136	87528384	21·0713075	7·6288837	·002252252
445	198025	88121125	21·0950231	7·6346067	·002247191
446	198916	88716536	21·1187121	7·6403213	·002242152
447	199809	89314623	21·1423745	7·6460272	·002237136
448	200704	89915392	21·1660105	7·6517247	·002232143
449	201601	90518849	21·1896201	7·6574138	·002227171
450	202500	91125000	21·2132034	7·6630943	·002222222
451	203401	91733851	21·2367606	7·6687665	·002217295
452	204304	92345408	21·2602916	7·6744303	·002212389
453	205209	92959677	21·2837967	7·6800857	·002207506
454	206116	93576664	21·3072758	7·6857328	·002202643
455	207025	94196375	21·3307290	7·6913717	·002197802
456	207936	94818816	21·3541565	7·6970023	·002192982
457	208849	95443993	21·3775583	7·7026246	·002188184
458	209764	96071912	21·4009346	7·7082388	·002183406
459	210681	96702579	21·4242853	7·7138448	·002178649
460	211600	97336000	21·4476106	7·7194426	·002173913
461	212521	97972181	21·4709106	7·7250325	·002169197
462	213444	98611128	21·4941853	7·7306141	·002164502
463	214369	99252847	21·5174348	7·7361877	·002159827
464	215296	99897344	21·5406592	7·7417532	·002155172
465	216225	100544625	21·5638587	7·7473109	·002150538
466	217156	101194696	21·5870331	7·7528606	·002145923
467	218089	101847563	21·6101828	7·7584023	·002141328
468	219024	102503232	21·6333077	7·7639361	·002136752
469	219961	103161709	21·6564078	7·7694620	·002132196
470	220900	103823000	21·6794834	7·7749801	·002127660
471	221841	104487111	21·7025344	7·7804904	·002123142
472	222784	105154048	21·7255610	7·7859928	·002118644
473	223729	105823817	21·7485632	7·7914875	·002114165
474	224676	106496424	21·7715411	7·7969745	·002109705
475	225625	107171875	21·7944947	7·8024538	·002105263
476	226576	107850176	21·8174242	7·8079254	·002100840
477	227529	108531333	21·8403297	7·8133892	·002096436
478	228484	109215352	21·8632111	7·8188456	·002092050
479	229441	109902239	21·8860686	7·8242942	·002087683
480	230400	110592000	21·9089023	7·8297353	·002083333
481	231361	111284641	21·9317122	7·8351688	·002079002
482	232324	111980168	21·9544984	7·8405949	·002074689
483	233289	112678587	21·9772610	7·8460134	·002070393
484	234256	113379904	22·0000000	7·8514244	·002066116
485	235225	114084125	22·0227155	7·8568281	·002061856
486	236196	114791256	22·0454077	7·8622242	·002057613

No.	Square	Cube	Square Root	Cube Root	Reciprocal
487	237169	115501303	22·0680765	7·8676130	·002053388
488	238144	116214272	22·0907220	7·8729944	·002049180
489	239121	116930169	22·1133444	7·8783684	·002044990
490	240100	117649000	22·1359436	7·8837352	·002040816
491	241081	118370771	22 1585198	7·8890946	·002036660
492	242064	119095488	22·1810730	7·8944468	·002032520
493	243049	119823157	22·2036033	7·8997917	·002028398
494	244036	120553784	22·2261108	7·9051294	·002024291
495	245025	121287375	22·2485955	7·9104599	·002020202
496	246016	122023936	22·2710575	7·9157832	·002016129
497	247009	122763473	22·2934968	7·9210994	·002012072
498	248004	123505992	22·3159136	7·9264085	·002008032
499	249001	124251499	22·3383079	7·9317104	·002004008
500	250000	125000000	22·3606798	7·9370053	·002000000
501	251001	125751501	22·3830293	7·9422931	·001996008
502	252004	126506008	22·4053565	7·9475739	·001992032
503	253009	127263527	22·4276615	7·9528477	·001988072
504	254016	128024064	22·4499443	7·9581144	·001984127
505	255025	128787625	22·4722051	7·9633743	·001980198
506	256036	129554216	22·4944438	7·9686271	·001976285
507	257049	130323843	22·5166605	7·9738731	·001972387
508	258064	131096512	22·5388553	7·9791122	·001968504
509	259081	131872229	22·5610283	7·9843444	·001964637
510	260100	132651000	22·5831796	7·9895697	·001960784
511	261121	133432831	22·6053091	7·9947883	·001956947
512	262144	134217728	22·6274170	8·0000000	·001953125
513	263169	135005697	22·6495033	8·0052049	·001949318
514	264196	135796744	22·6715681	8·0104032	·001945525
515	265225	136590875	22·6936114	8·0155946	·001941748
516	266256	137388096	22·7156334	8·0207794	·001937984
517	267289	138188413	22·7376340	8·0259574	·001934236
518	268324	138991832	22·7596134	8·0311287	·001930502
519	269361	139798359	22·7815715	8·0362935	·001926782
520	270400	140608000	22·8035085	8·0414515	·001923077
521	271441	141420761	22·8254244	8·0466030	·001919386
522	272484	142236648	22·8473193	8·0517479	·001915709
523	273529	143055667	22·8691933	8·0568862	·001912046
524	274576	143877824	22·8910463	8·0620180	·001908397
525	275625	144703125	22·9128785	8·0671432	·001904762
526	276676	145531576	22·9346899	8·0722620	·001901141
527	277729	146363183	22·9564806	8·0773743	·001897533
528	278784	147197952	22·9782506	8·0824800	·001893939
529	279841	148035889	23·0000000	8·0875794	·001890359
530	280900	148877000	23·0217289	8·0926723	·001886792
531	281961	149721291	23·0434372	8·0977589	·001883239
532	283024	150568768	23·0651252	8·1028390	·001879699
533	284089	151419437	23·0867928	8·1079128	·001876173
534	285156	152273304	23·1084400	8·1129803	·001872659
535	286225	153130375	23·1300670	8·1180414	·001869159

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536	287296	153990656	23·1516738	8·1230962	·001865672
537	288369	154854153	23·1732605	8·1281447	·001862197
538	289444	155720872	23·1948270	8·1331870	·001858736
539	290521	156590819	23·2163735	8·1382230	·001855288
540	291600	157464000	23·2379001	8·1432529	·001851852
541	292681	158340421	23·2594067	8·1482765	·001848429
542	293764	159220088	23·2808935	8·1532939	·001845018
543	294849	160103007	23·3023604	8·1583051	·001841621
544	295936	160989184	23·3238076	8·1633102	·001838235
545	297025	161878625	23·3452351	8·1683092	·001834862
546	298116	162771336	23·3666429	8·1733020	·001831502
547	299209	163667323	23·3880311	8·1782888	·001828154
548	300304	164566592	23·4093998	8·1832695	·001824818
549	301401	165469149	23·4307490	8·1882441	·001821494
550	302500	166375000	23·4520788	8·1932127	·001818182
551	303601	167284151	23·4733892	8·1981753	·001814882
552	304704	168196608	23·4946802	8·2031319	·001811594
553	305809	169112377	23·5159520	8·2080825	·001808318
554	306916	170031464	23·5372046	8·2130271	·001805054
555	308025	170953875	23·5584380	8·2179657	·001801802
556	309136	171879616	23·5796522	8·2228985	·001798561
557	310249	172808693	23·6008474	8·2278254	·001795332
558	311364	173741112	23·6220236	8·2327463	·001792115
559	312481	174676879	23·6431808	8·2376614	·001788909
560	313600	175616000	23·6643191	8·2425706	·001785714
561	314721	176558481	23·6854386	8·2474740	·001782531
562	315844	177504328	23·7065392	8·2523715	·001779359
563	316969	178453547	23·7276210	8·2572633	·001776199
564	318096	179406144	23·7486842	8·2621492	·001773050
565	319225	180362125	23·7697286	8·2670294	·001769912
566	320356	181321496	23·7907545	8·2719039	·001766784
567	321489	182284263	23·8117618	8·2767726	·001763668
568	322624	183250432	23·8327506	8·2816355	·001760563
569	323761	184220009	23·8537209	8·2864928	·001757469
570	324900	185193000	23·8746728	8·2913444	·001754386
571	326041	186169411	23·8956063	8·2961903	·001751313
572	327184	187149248	23·9165215	8·3010304	·001748252
573	328329	188132517	23·9374184	8·3058651	·001745201
574	329476	189119224	23·9582971	8·3106941	·001742160
575	330625	190109375	23·9791576	8·3155175	·001739130
576	331776	191102976	24·0000000	8·3203353	·001736111
577	332929	192100033	24·0208243	8·3251475	·001733102
578	334084	193100552	24·0416306	8·3299542	·001730104
579	335241	194104539	24·0624188	8·3347553	·001727116
580	336400	195112000	24·0831891	8·3395509	·001724138
581	337561	196122941	24·1039416	8·3443410	·001721170
582	338724	197137368	24·1246762	8·3491256	·001718213
583	339889	198155287	24·1453929	8·3539047	·001715266
584	341056	199176704	24·1660919	8·3586784	·001712329

No.	Square	Cube	Square Root	Cube Root	Reciprocal
585	342225	200201625	24·1867732	8·3634466	·001709402
586	343396	201230056	24·2074369	8·3682095	·001706485
587	344569	202262003	24·2280829	8·3729668	·001703578
588	345744	203297472	24·2487113	8·3777188	·005700680
589	346921	204336469	24·2693222	8·3824653	·001697793
590	348100	205379000	24·2899156	8·3872065	·001694915
591	349281	206425071	24·3104916	8·3919423	·001692047
592	350464	207474688	24·3310501	8·3966729	·001689189
593	351649	208527857	24·3515913	8·4013981	·001686341
594	352836	209584584	24·3721152	8·4061180	·001683502
595	354025	210644875	24·3926218	8·4108326	·001680672
596	355216	211708736	24·4131112	8·4155419	·001677852
597	356409	212776173	24·4335834	8·4202460	·001675042
598	357604	213847192	24·4540385	8·4249448	·001672241
599	358801	214921799	24·4744765	8·4296383	·001669449
600	360000	216000000	24·4948974	8·4343267	·001666667
601	361201	217081801	24·5153013	8·4390098	·001663894
602	362404	218167208	24·5356883	8·4436877	·001661130
603	363609	219256227	24·5560583	8·4483605	·001658375
604	364816	220348864	24·5764115	8·4530281	·001655629
605	366025	221445125	24·5967478	8·4576906	·001652893
606	367236	222545016	24·6170673	8·4623479	·001650165
607	368449	223648543	24·6373700	8·4670000	·001647446
608	369664	224755712	24·6576560	8·4716471	·001644737
609	370881	225866529	24·6779254	8·4762892	·001642036
610	372100	226981000	24·6981781	8·4809261	·001639344
611	373321	228099131	24·7184142	8·4855579	·001636661
612	374544	229220928	24·7386338	8·4901848	·001633987
613	375769	230346397	24·7588368	8·4948065	·001631321
614	376996	231475544	24·7790234	8·4994233	·001628664
615	378225	232608375	24·7991935	8·5040350	·001626016
616	379456	233744896	24·8193473	8·5086417	·001623377
617	380689	234885113	24·8394847	8·5132435	·001620746
618	381924	236029032	24·8596058	8·5178403	·001618123
619	383161	237176659	24·8797106	8·5224321	·001615509
620	384400	238328000	24·8997992	8·5270189	·001612903
621	385641	239483061	24·9198716	8·5316009	·001610306
622	386884	240641848	24·9399278	8·5361780	·001607717
623	388129	241804367	24·9599679	8·5407501	·001605136
624	389376	242970624	24·9799920	8·5453173	·001602564
625	390625	244140625	25·0000000	8·5498797	·001600000
626	391876	245314376	25·0199920	8·5544372	·001597444
627	393129	246491883	25·0399681	8·5589899	·001594896
628	394384	247673152	25·0599282	8·5635377	·001592357
629	395641	248858189	25·0798724	8·5680807	·001589825
630	396900	250047000	25·0998008	8·5726189	·001587302
631	398161	251239591	25·1197134	8·5771523	·001584786
632	399424	252435968	25·1396102	8·5816809	·001582278
633	400689	253636137	25·1594913	8·5862047	·001579779

No.	Square	Cube	Square Root	Cube Root	Reciprocal
634	401956	254840104	25·1793566	8·5907238	·001577287
635	403225	256047875	25·1992063	8·5952380	·001574803
636	404496	257259456	25·2190404	8·5997476	·001572327
637	405769	258474853	25·2388589	8·6042525	·001569859
638	407044	259694072	25·2586619	8·6087526	·001567398
639	408321	260917119	25·2784493	8·6132480	·001564945
640	409600	262144000	25·2982213	8·6177388	·001562500
641	410881	263374721	25·3179778	8·6222248	·001560062
642	412164	264609288	25·3377189	8·6267063	·001557632
643	413449	265847707	25·3574447	8·6311830	·001555210
644	414736	267089984	25·3771551	8·6356551	·001552795
645	416025	268336125	25·3968502	8·6401226	·001550388
646	417316	269586136	25·4165301	8·6445855	·001547988
647	418609	270840023	25·4361947	8·6490437	·001545595
648	419904	272097792	25·4558441	8·6534974	·001543210
649	421201	273359449	25·4754784	8·6579465	·001540832
650	422500	274625000	25·4950976	8·6623911	·001538462
651	423801	275894451	25·5147016	8·6668310	·001536098
652	425104	277167808	25·5342907	8·6712665	·001533742
653	426409	278445077	25·5538647	8·6756974	·001531394
654	427716	279726264	25·5734237	8·6801237	·001529052
655	429025	281011375	25·5929678	8·6845456	·001526718
656	430336	282300416	25·6124969	8·6889630	·001524390
657	431649	283593393	25·6320112	8·6933759	·001522070
658	432964	284890312	25·6515107	8·6977843	·001519757
659	434281	286191179	25·6709953	8·7021882	·001517451
660	435600	287496000	25·6904652	8·7065877	·001515152
661	436921	288804781	25·7099203	8·7109827	·001512859
662	438244	290117528	25·7293607	8·7153734	·001510574
663	439569	291434247	25·7487864	8·7197596	·001508296
664	440896	292754944	25·7681975	8·7241414	·001506024
665	442225	294079625	25·7875939	8·7285187	·001503759
666	443556	295408296	25·8069758	8·7328918	·001501502
667	444889	296740963	25·8263431	8·7372604	·001499250
668	446224	298077632	25·8456960	8·7416246	·001497006
669	447561	299418309	25·8650343	8·7459846	·001494768
670	448900	300763000	25·8843582	8·7503401	·001492537
671	450241	302111711	25·9036677	8·7546913	·001490313
672	451584	303464448	25·9229628	8·7590383	·001488095
673	452929	304821217	25·9422435	8·7633809	·001485884
674	454276	306182024	25·9615100	8·7677192	·001483680
675	455625	307546875	25·9807621	8·7720532	·001481481
676	456976	308915776	26·0000000	8·7763830	·001479290
677	458329	310288733	26·0192237	8·7807084	·001477105
678	459684	311665752	26·0384331	8·7850296	·001474926
679	461041	313046839	26·0576284	8·7893466	·001472754
680	462400	314432000	26·0768096	8·7936593	·001470588
681	463761	315821241	26·0959767	8·7979679	·001468429
682	465124	317214568	26·1151297	8·8022721	·001466276

No.	Square	Cube	Square Root	Cube Root	Reciprocal
683	466489	318611987	26·1342687	8·8065722	·001464129
684	467856	320013504	26·1533937	8·8108681	·001461988
685	469225	321419125	26·1725047	8·8151598	·001459854
686	470596	322828856	26·1916017	8·8194474	·001457726
687	471969	324242703	26·2106848	8·8237307	·001455604
688	473344	325660672	26·2297541	8·8280099	·001453488
689	474721	327082769	26·2488095	8·8322850	·001451379
690	476100	328509000	26·2678511	8·8365559	·001449275
691	477481	329939371	26·2868789	8·8408227	·001447178
692	478864	331373888	26·3058929	8·8450854	·001445087
693	480249	332812557	26·3248932	8·8493440	·001443001
694	481636	334255384	26·3438797	8·8535985	·001440922
695	483025	335702375	26·3628527	8·8578489	·001438849
696	484416	337153536	26·3818119	8·8620952	·001436782
697	485809	338608873	26·4007576	8·8663375	·001434720
698	487204	340068392	26·4196896	8·8705757	·001432665
699	488601	341532099	26·4386081	8·8748099	·001430615
700	490000	343000000	26·4575131	8·8790400	·001428571
701	491401	344472101	26·4764046	8·8832661	·001426534
702	492804	345948408	26·4952826	8·8874882	·001424501
703	494209	347428927	26·5141472	8·8917063	·001422475
704	495616	348913664	26·5329983	8·8959204	·001420455
705	497025	350402625	26·5518361	8·9001304	·001418440
706	498436	351895816	26·5706605	8·9043366	·001416431
707	499849	353393243	26·5894716	8·9085387	·001414427
708	501264	354894912	26·6082694	8·9127369	·001412429
709	502681	356400829	26·6270539	8·9169311	·001410437
710	504100	357911000	26·6458252	8·9211214	·001408451
711	505521	359425431	26·6645833	8·9253078	·001406470
712	506944	360944128	26·6833281	8·9294902	·001404494
713	508369	362467097	26·7020598	8·9336687	·001402525
714	509796	363994344	26·7207784	8·9378433	·001400560
715	511225	365525875	26·7394839	8·9420140	·001398601
716	512656	367061696	26·7581763	8·9461809	·001396648
717	514089	368601813	26·7768557	8·9503438	·001394700
718	515524	370146232	26·7955220	8·9545029	·001392758
719	516961	371694959	26·8141754	8·9586581	·001390821
720	518400	373248000	26·8328157	8·9628095	·001388889
721	519841	374805361	26·8514432	8·9669570	·001386963
722	521284	376367048	26·8700577	8·9711007	·001385042
723	522729	377933067	26·8886593	8·9752406	·001383126
724	524176	379503424	26·9072481	8·9793766	·001381215
725	525625	381078125	26·9258240	8·9835089	·001379310
726	527076	382657176	26·9443872	8·9876373	·001377410
727	528529	384240583	26·9629375	8·9917620	·001375516
728	529984	385828352	26·9814751	8·9958829	·001373626
729	531441	387420489	27·0000000	9·0000000	·001371742
730	532900	389017000	27·0185122	9·0041134	·001369863
731	534361	390617891	27·0370117	9·0082229	·001367989

No.	Square	Cube	Square Root	Cube Root	Reciprocal
732	535824	392223168	27·0554985	9·0123288	·001366120
733	537289	393832837	27·0739727	9·0164309	·001364256
734	538756	395446904	27·0924344	9·0205293	·001362398
735	540225	397065375	27·1108834	9·0246239	·001360544
736	541696	398688256	27·1293199	9·0287149	·001358696
737	543169	400315553	27·1477439	9·0328021	·001356852
738	544644	401947272	27·1661554	9·0368857	·001355014
739	546121	403583419	27·1845544	9·0409655	·001353180
740	547600	405224000	27·2029410	9·0450417	·001351351
741	549081	406869021	27·2213152	9·0491142	·001349528
742	550564	408518488	27·2396769	9·0531831	·001347709
743	552049	410172407	27·2580263	9·0572482	·001345895
744	553536	411830784	27·2763634	9·0613098	·001344086
745	555025	413493625	27·2946881	9·0653677	·001342282
746	556516	415160936	27·3130006	9·0694220	·001340483
747	558009	416832723	27·3313007	9·0734726	·001338688
748	559504	418508992	27·3495887	9·0775197	·001336898
749	561001	420189749	27·3678644	9·0815631	·001335113
750	562500	421875000	27·3861279	9·0856030	·001333333
751	564001	423564751	27·4043792	9·0896392	·001331558
752	565504	425259008	27·4226184	9·0936719	·001329787
753	567009	426957777	27·4408455	9·0977010	·001328021
754	568516	428661064	27·4590604	9·1017265	·001326260
755	570025	430368875	27·4772633	9·1057485	·001324503
756	571536	432081216	27·4954542	9·1097669	·001322751
757	573049	433798093	27·5136330	9·1137818	·001321004
758	574564	435519512	27·5317998	9·1177931	·001319261
759	576081	437245479	27·5499546	9·1218010	·001317523
760	577600	438976000	27·5680975	9·1258053	·001315789
761	579121	440711081	27·5862284	9·1298061	·001314060
762	580644	442450728	27·6043475	9·1338034	·001312336
763	582169	444194947	27·6224546	9·1377971	·001310616
764	583696	445943744	27·6405499	9·1417874	·001308901
765	585225	447697125	27·6586334	9·1457742	·001307190
766	586756	449455096	27·6767050	9·1497576	·001305483
767	588289	451217663	27·6947648	9·1537375	·001303781
768	589824	452984832	27·7128129	9·1577139	·001302083
769	591361	454756609	27·7308492	9·1616869	·001300390
770	592900	456533000	27·7488739	9·1656565	·001298701
771	594441	458314011	27·7668868	9·1696225	·001297017
772	595984	460099648	27·7848880	9·1735852	·001295337
773	597529	461889917	27·8028775	9·1775445	·001293661
774	599076	463684824	27·8208555	9·1815003	·001291990
775	600625	465484375	27·8388218	9·1854527	·001290323
776	602176	467288576	27·8567766	9·1894018	·001288660
777	603729	469097433	27·8747197	9·1933474	·001287001
778	605284	470910952	27·8926514	9·1972897	·001285347
779	606841	472729139	27·9105715	9·2012286	·001283697
780	608400	474552000	27·9284801	9·2051641	·001282051

No.	Square	Cube	Square Root	Cube Root	Reciprocal
781	609961	476379541	27·9463772	9·2090962	·001280410
782	611524	478211768	27·9642629	9·2130250	·001278772
783	613089	480048687	27·9821372	9·2169505	·001277139
784	614656	481890304	28·0000000	9·2208726	·001275510
785	616225	483736625	28·0178515	9·2247914	·001273885
786	617796	485587656	28·0356915	9·2287068	·001272265
787	619369	487443403	28·0535203	9·2326189	·001270648
788	620944	489303872	28·0713377	9·2365277	·001269036
789	622521	491169069	28·0891438	9·2404333	·001267427
790	624100	493039000	28·1069386	9·2443355	·001265823
791	625681	494913671	28·1247222	9·2482344	·001264223
792	627264	496793088	28·1424946	9·2521300	·001262626
793	628849	498677257	28·1602557	9·2560224	·001261034
794	630436	500566184	28·1780056	9·2599114	·001259446
795	632025	502459875	28·1957444	9·2637973	·001257862
796	633616	504358336	28·2134720	9·2676798	·001256281
797	635209	506261573	28·2311884	9·2715592	·001254705
798	636804	508169592	28·2488938	9·2754352	·001253133
799	638401	510082399	28·2665881	9·2793081	·001251564
800	640000	512000000	28·2842712	9·2831777	·001250000
801	641601	513922401	28·3019434	9·2870440	·001248439
802	643204	515849608	28·3196045	9·2909072	·001246883
803	644809	517781627	28·3372546	9·2947671	·001245330
804	646416	519718464	28·3548938	9·2986239	·001243781
805	648025	521660125	28·3725219	9·3024775	·001242236
806	649636	523606616	28·3901391	9·3063278	·001240695
807	651249	525557943	28·4077454	9·3101750	·001239157
808	652864	527514112	28·4253408	9·3140190	·001237624
809	654481	529475129	28·4429253	9·3178599	·001236094
810	656100	531441000	28·4604989	9·3216975	·001234568
811	657721	533411731	28·4780617	9·3255320	·001233046
812	659344	535387328	28·4956137	9·3293634	·001231527
813	660969	537367797	28·5131549	9·3331916	·001230012
814	662596	539353144	28·5306852	9·3370167	·001228501
815	664225	541343375	28·5482048	9·3408386	·001226994
816	665856	543338496	28·5657137	9·3446575	·001225490
817	667489	545338513	28·5832119	9·3484731	·001223990
818	669124	547343432	28·6006993	9·3522857	·001222494
819	670761	549353259	28·6181760	9·3560952	·001221001
820	672400	551368000	28·6356421	9·3599016	·001219512
821	674041	553387661	28·6530976	9·3637049	·001218027
822	675684	555412248	28·6705424	9·3675051	·001216545
823	677329	557441767	28·6879766	9·3713022	·001215067
824	678976	559476224	28·7054002	9·3750963	·001213592
825	680625	561515625	28·7228132	9·3788873	·001212121
826	682276	563559976	28·7402157	9·3826752	·001210654
827	683929	565609283	28·7576077	9·3864600	·001209190
828	685584	567663552	28·7749891	9·3902419	·001207729
829	687241	569722789	28·7923601	9·3940206	·001206272

No.	Square	Cube	Square Root	Cube Root	Reciprocal
830	688900	571787000	28·8097206	9·3977964	·001204819
831	690561	573856191	28·8270706	9·4015691	·001203369
832	692224	575930368	28·8444102	9·4053387	·001201923
833	693889	578009537	28·8617394	9·4091054	·001200480
834	695556	580093704	28·8790582	9·4128690	·001199041
835	697225	582182875	28·8963666	9·4166297	·001197605
836	698896	584277056	28·9136646	9·4203873	·001196172
837	700569	586376253	28·9309523	9·4241420	·001194743
838	702244	588480472	28·9482297	9·4278936	·001193317
839	703921	590589719	28·9654967	9·4316423	·001191895
840	705600	592704000	28·9827535	9·4353880	·001190476
841	707281	594823321	29·0000000	9·4391307	·001189061
842	708964	596947688	29·0172363	9·4428704	·001187648
843	710649	599077107	29·0344623	9·4466072	·001186240
844	712336	601211584	29·0516781	9·4503410	·001184834
845	714025	603351125	29·0688837	9·4540719	·001183432
846	715716	605495736	29·0860791	9·4577999	·001182033
847	717409	607645423	29·1032644	9·4615249	·001180638
848	719104	609800192	29·1204396	9·4652470	·001179245
849	720801	611960049	29·1376046	9·4689661	·001177856
850	722500	614125000	29·1547595	9·4726824	·001176471
851	724201	616295051	29·1719043	9·4763957	·001175088
852	725904	618470208	29·1890390	9·4801061	·001173709
853	727609	620650477	29·2061637	9·4838136	·001172333
854	729316	622835864	29·2232784	9·4875182	·001170960
855	731025	625026375	29·2403830	9·4912200	·001169591
856	732736	627222016	29·2574777	9·4949188	·001168224
857	734449	629422793	29·2745623	9·4986147	·001166861
858	736164	631628712	29·2916370	9·5023078	·001165501
859	737881	633839779	29·3087018	9·5059980	·001164144
860	739600	636056000	29·3257566	9·5096854	·001162791
861	741321	638277381	29·3428015	9·5133699	·001161440
862	743044	640503928	29·3598365	9·5170515	·001160093
863	744769	642735647	29·3768616	9·5207303	·001158749
864	746496	644972544	29·3938769	9·5244063	·001157407
865	748225	647214625	29·4108823	9·5280794	·001156069
866	749956	649461896	29·4278779	9·5317497	·001154734
867	751689	651714363	29·4448637	9·5354172	·001153403
868	753424	653972032	29·4618397	9·5390818	·001152074
869	755161	656234909	29·4788059	9·5427437	·001150748
870	756900	658503000	29·4957624	9·5464027	·001149425
871	758641	660776311	29·5127091	9·5500589	·001148106
872	760384	663054848	29·5296461	9·5537123	·001146789
873	762129	665338617	29·5465734	9·5573630	·001145475
874	763876	667627624	29·5634910	9·5610108	·001144165
875	765625	669921875	29·5803989	9·5646559	·001142857
876	767376	672221376	29·5972972	9·5682982	·001141553
877	769129	674526133	29·6141858	9·5719377	·001140251
878	770884	676836152	29·6310648	9·5755745	·001138952

No.	Square	Cube	Square Root	Cube Root	Reciprocal
879	772641	679151439	29·6479342	9·5792085	·001137656
880	774400	681472000	29·6647939	9·5828397	·001136364
881	776161	683797841	29·6816442	9·5864682	·001135074
882	777924	686128968	29·6984848	9·5900939	·001133787
883	779689	688465387	29·7153159	9·5937169	·001132503
884	781456	690807104	29·7321375	9·5973373	·001131222
885	783225	693154125	29·7489496	9·6009548	·001129944
886	784996	695506456	29·7657521	9·6045696	·001128668
887	786769	697864103	29·7825452	9·6081817	·001127396
888	788544	700227072	29·7993289	9·6117911	·001126126
889	790321	702595369	29·8161030	9·6153977	·001124859
890	792100	704969000	29·8328678	9·6190017	·001123596
891	793881	707347971	29·8496231	9·6226030	·001122334
892	795664	709732288	29·8663690	9·6262016	·001121076
893	797449	712121957	29·8831056	9·6297975	·001119821
894	799236	714516984	29·8998328	9·6333907	·001118568
895	801025	716917375	29·9165506	9·6369812	·001117318
896	802816	719323136	29·9332591	9·6405690	·001116071
897	804609	721734273	29·9499583	9·6441542	·001114827
898	806404	724150792	29·9666481	9·6477367	·001113586
899	808201	726572699	29·9833287	9·6515166	·001112347
900	810000	729000000	30·0000000	9·6548938	·001111111
901	811801	731432701	30·0166620	9·6584684	·001109878
902	813604	733870808	30·0333148	9·6620403	·001108647
903	815409	736314327	30·0499584	9·6656096	·001107420
904	817216	738763264	30·0665928	9·6691762	·001106195
905	819025	741217625	30·0832179	9·6727403	·001104972
906	820836	743677416	30·0998339	9·6763017	·001103753
907	822649	746142643	30·1164407	9·6798604	·001102536
908	824464	748613312	30·1330383	9·6834166	·001101322
909	826281	751089429	30·1496269	9·6869701	·001100110
910	828100	753571000	30·1662063	9·6905211	·001098901
911	829921	756058031	30·1827765	9·6940694	·001097695
912	831744	758550528	30·1993377	9·6976151	·001096491
913	833569	761048497	30·2158899	9·7011583	·001095290
914	835396	763551944	30·2324329	9·7046989	·001094092
915	837225	766060875	30·2489669	9·7082369	·001092896
916	839056	768575296	30·2654919	9·7117723	·001091703
917	840889	771095213	30·2820079	9·7153051	·001090513
918	842724	773620632	30·2985148	9·7188354	·001089325
919	844561	776151559	30·3150128	9·7223631	·001088139
920	846400	778688000	30·3315018	9·7258883	·001086957
921	848241	781229961	30·3479818	9·7294109	·001085776
922	850084	783777448	30·3644529	9·7329309	·001084599
923	851929	786330467	30·3809151	9·7364484	·001083424
924	853776	788889024	30·3973683	9·7399634	·001082251
925	855625	791453125	30·4138127	9·7434758	·001081081
926	857476	794022776	30·4302481	9·7469857	·001079914
927	859329	796597983	30·4466747	9·7504930	·001078749

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No.	Square	Cube	Square Root	Cube Root	Reciprocal
928	861184	799178752	30·4630924	9·7539979	·001077586
929	863041	801765089	30·4795013	9·7575002	·001076426
930	864900	804357000	30·4959014	9·7610001	·001075269
931	866761	806954491	30·5122926	9·7644974	·001074114
932	868624	809557568	30·5286750	9·7679922	·001072961
933	870489	812166237	30·5450487	9·7714845	·001071811
934	872356	814780504	30·5614136	9·7749743	·001070664
935	874225	817400375	30·5777697	9·7784616	·001069519
936	876096	820025856	30·5941171	9·7819466	·001068376
937	877969	822656953	30·6104557	9·7854288	·001067236
938	879844	825293672	30·6267857	9·7889087	·001066098
939	881721	827936019	30·6431069	9·7923861	·001064963
940	883600	830584000	30·6594194	9·7958611	·001063830
941	885481	833237621	30·6757233	9·7993336	·001062699
942	887364	835896888	30·6920185	9·8028036	·001061571
943	889249	838561807	30·7083051	9·8062711	·001060445
944	891136	841232384	30·7245830	9·8097362	·001059322
945	893025	843908625	30·7408523	9·8131989	·001058201
946	894916	846590536	30·7571130	9·8166591	·001057082
947	896809	849278123	30·7733651	9·8201169	·001055966
948	898704	851971392	30·7896086	9·8235723	·001054852
949	900601	854670349	30·8058436	9·8270252	·001053741
950	902500	857375000	30·8220700	9·8304757	·001052632
951	904401	860085351	30·8382879	9·8339238	·001051525
952	906304	862801408	30·8544972	9·8373695	·001050420
953	908209	865523177	30·8706981	9·8408127	·001049318
954	910116	868250664	30·8868904	9·8442536	·001048218
955	912025	870983875	30·9030743	9·8476920	·001047120
956	913936	873722816	30·9192497	9·8511280	·001046025
957	915849	876467493	30·9354166	9·8545617	·001044932
958	917764	879217912	30·9515751	9·8579929	·001043841
959	919681	881974079	30·9677251	9·8614218	·001042753
960	921600	884736000	30·9838668	9·8648483	·001041667
961	923521	887503681	31·0000000	9·8682724	·001040583
962	925444	890277128	31·0161248	9·8716941	·001039501
963	927369	893056347	31·0322413	9·8751135	·001038422
964	929296	895841344	31·0483494	9·8785305	·001037344
965	931225	898632125	31·0644491	9·8819451	·001036269
966	933156	901428696	31·0805405	9·8853574	·001035197
967	935089	904231063	31·0966236	9·8887673	·001034126
968	937024	907039232	31·1126984	9·8921749	·001033058
969	938961	909853209	31·1287648	9·8955801	·001031992
970	940900	912673000	31·1448230	9·8989830	·001030928
971	942841	915498611	31·1608729	9·9023835	·001029866
972	944784	918330048	31·1769145	9·9057817	·001028807
973	946729	921167317	31·1929479	9·9091776	·001027749
974	948676	924010424	31·2089731	9·9125712	·001026694
975	950625	926859375	31·2249900	9·9159624	·001025641
976	952576	929714176	31·2409987	9·9193513	·001024590

No.	Square	Cube	Square Root	Cube Root	Reciprocal
977	954529	932574833	31·2569992	9·9227879	·001023541
978	956484	935441352	31·2729915	9·9261222	·001022495
979	958441	938313739	31·2889757	9·9295042	·001021450
980	960400	941192000	31·3049517	9·9328839	·001020408
981	962361	944076141	31·3209195	9·9362613	·001019368
982	964324	946966168	31·3368792	9·9396363	·001018330
983	966289	949862087	31·3528308	9·9430092	·001017294
984	968256	952763904	31·3687743	9·9463797	·001016260
985	970225	955671625	31·3847097	9·9497479	·001015228
986	972196	958585256	31·4006369	9·9531138	·001014199
987	974169	961504803	31·4165561	9·9564775	·001013171
988	976144	964430272	31·4324673	9·9598389	·001012146
989	978121	967361669	31·4483704	9·9631981	·001011122
990	980100	970299000	31·4642654	9·9665549	·001010101
991	982081	973242271	31·4801525	9·9699095	·001009082
992	984064	976191488	31·4960315	9·9732619	·001008065
993	986049	979146657	31·5119025	9·9766120	·001007049
994	988036	982107784	31·5277655	9·9799599	·001006036
995	990025	985074875	31·5436206	9·9833055	·001005025
996	992016	988047936	31·5594677	9·9866488	·001004016
997	994009	991026973	31·5753068	9·9899900	·001003009
998	996004	994011992	31·5911380	9·9933289	·001002004
999	998001	997002999	31·6069613	9·9966656	·001001001
1000	1000000	1000000000	31·6227766	10·0000000	·0010000000
1001	1002001	1003003001	31·6385840	10·0033322	·0009990010
1002	1004004	1006012008	31·6543836	10·0066622	·0009980040
1003	1006009	1009027027	31·6701752	10·0099899	·0009970090
1004	1008016	1012048064	31·6859590	10·0133155	·0009960159
1005	1010025	1015075125	31·7017349	10·0166389	·0009950249
1006	1012036	1018108216	31·7175030	10·0199601	·0009940358
1007	1014049	1021147343	31·7332633	10·0232791	·0009930487
1008	1016064	1024192512	31·7490157	10·0265958	·0009920635
1009	1018081	1027243729	31·7647603	10·0299104	·0009910803
1010	1020100	1030301000	31·7804972	10·0332228	·0009900990
1011	1022121	1033364331	31·7962262	10·0365330	·0009891197
1012	1024144	1036433728	31·8119474	10·0398410	·0009881423
1013	1026169	1039509197	31·8276609	10·0431469	·0009871668
1014	1028196	1042590744	31·8433666	10·0464506	·0009861933
1015	1030225	1045678375	31·8590646	10·0497521	·0009852217
1016	1032256	1048772096	31·8747549	10·0530514	·0009842520
1017	1034289	1051871913	31·8904374	10·0563485	·0009832842
1018	1036324	1054977832	31·9061123	10·0596435	·0009823183
1019	1038361	1058089859	31·9217794	10·0629364	·0009813543
1020	1040400	1061208000	31·9374388	10·0662271	·0009808922
1021	1042441	1064332261	31·9530906	10·0695156	·0009794319
1022	1044484	1067462648	31·9687347	10·0728020	·0009784736
1023	1046529	1070599167	31·9843712	10·0760863	·0009775171
1024	1048576	1073741824	32·0000000	10·0793684	·0009765625
1025	1050625	1076890625	32·0156212	10·0826484	·0009756098

No.	Square	Cube	Square Root	Cube Root	Reciprocal
1026	1052676	1080045576	32·0312348	10·0859262	·0009746589
1027	1054729	1083206683	32·0468407	10·0892019	·0009737098
1028	1056784	1086373952	32·0624391	10·0924755	·0009727626
1029	1058841	1089547389	32·0780298	10·0957469	·0009718173
1030	1060900	1092727000	32·0936131	10·0990163	·0009708738
1031	1062961	1095912791	32·1091887	10·1022835	·0009699321
1032	1065024	1099104768	32·1247568	10·1055487	·0009689922
1033	1067089	1102302937	32·1403173	10·1088117	·0009680542
1034	1069156	1105507304	32·1558704	10·1120726	·0009671180
1035	1071225	1108717875	32·1714159	10·1153314	·0009661836
1036	1073296	1111934656	32·1869539	10·1185882	·0009652510
1037	1075369	1115157653	32·2024844	10·1218428	·0009643202
1038	1077444	1118386872	32·2180074	10·1250953	·0009633911
1039	1079521	1121622319	32·2335229	10·1283457	·0009624639
1040	1081600	1124864000	32·2490310	10·1315941	·0009615385
1041	1083681	1128111921	32·2645316	10·1348403	·0009606148
1042	1085764	1131366088	32·2800248	10·1380845	·0009596929
1043	1087849	1134626507	32·2955105	10·1413266	·0009587728
1044	1089936	1137893184	32·3109888	10·1445667	·0009578544
1045	1092025	1141166125	32·3264598	10·1478047	·0009569378
1046	1094116	1144445336	32·3419233	10·1510406	·0009560229
1047	1096209	1147730823	32·3573794	10·1542744	·0009551098
1048	1098304	1151022592	32·3728281	10·1575062	·0009541985
1049	1100401	1154320649	32·3882695	10·1607359	·0009532888
1050	1102500	1157625000	32·4037035	10·1639636	·0009523810
1051	1104601	1160935651	32·4191301	10·1671893	·0009514748
1052	1106704	1164252608	32·4345495	10·1704129	·0009505703
1053	1108809	1167575877	32·4499615	10·1736344	·0009496676
1054	1110916	1170905464	32·4653662	10·1768539	·0009487666
1055	1113025	1174241375	32·4807635	10·1800714	·0009478673
1056	1115136	1177583616	32·4961536	10·1832868	·0009469697
1057	1117249	1180932193	32·5115364	10·1865002	·0009460738
1058	1119364	1184287112	32·5269119	10·1897116	·0009451796
1059	1121481	1187648379	32·5422802	10·1929209	·0009442871
1060	1123600	1191016000	32·5576412	10·1961283	·0009433962
1061	1125721	1194389981	32·5729949	10·1993336	·0009425071
1062	1127844	1197770328	32·5883415	10·2025369	·0009416196
1063	1129969	1201157047	32·6036807	10·2057382	·0009407338
1064	1132096	1204550144	32·6190129	10·2089375	·0009398496
1065	1134225	1207949625	32·6343377	10·2121347	·0009389671
1066	1136356	1211355496	32·6496554	10·2153300	·0009380863
1067	1138489	1214767763	32·6649659	10·2185233	·0009372071
1068	1140624	1218186432	32·6802693	10·2217146	·0009363296
1069	1142761	1221611509	32·6955654	10·2249039	·0009354537
1070	1144900	1225043000	32·7108544	10·2280912	·0009345794
1071	1147041	1228480911	32·7261363	10·2312766	·0009337068
1072	1149184	1231925248	32·7414111	10·2344599	·0000328358
1073	1151329	1235376017	32·7566787	10·2376413	·0009319664
1074	1153476	1238833224	32·7719392	10·2408207	·0009310987

No.	Square	Cube	Square Root	Cube Root	Reciprocal
1075	1155625	1242296875	32·7871926	10·2439981	·0009302326
1076	1157776	1245766976	32·8024389	10·2471735	·0009293680
1077	1159929	1249243533	32·8176782	10·2503470	·0009285051
1078	1162084	1252726552	32·8329103	10·2535186	·0009276438
1079	1164241	1256216039	32·8481354	10·2566881	·0009267841
1080	1166400	1259712000	32·8633535	10·2598557	·0009259259
1081	1168561	1263214441	32·8785644	10·2630213	·0009250694
1082	1170724	1266723368	32·8937684	10·2661850	·0009242144
1083	1172889	1270238787	32·9089653	10·2693467	·0009233610
1084	1175056	1273760704	32·9241553	10·2725065	·0009225092
1085	1177225	1277289125	32·9393382	10·2756644	·0009216590
1086	1179396	1280824056	32·9545141	10·2788203	·0009208103
1087	1181569	1284365503	32·9696830	10·2819743	·0009199632
1088	1183744	1287913472	32·9848450	10·2851264	·0009191176
1089	1185921	1291467969	33·0000000	10·2882765	·0009182736
1090	1188100	1295029000	33·0151480	10·2914247	·0009174312
1091	1190281	1298596571	33·0302891	10·2945709	·0009165903
1092	1192464	1302170688	33·0454233	10·2977153	·0009157509
1093	1194649	1305751357	33·0605505	10·3008577	·0009149131
1094	1196836	1309338584	33·0756708	10·3039982	·0009140768
1095	1199025	1312932375	33·0907842	10·3071368	·0009132420
1096	1201216	1316532736	33·1058907	10·3102735	·0009124088
1097	1203409	1320139673	33·1209903	10·3134083	·0009115770
1098	1205604	1323753192	33·1360830	10·3165411	·0009107468
1099	1207801	1327373299	33·1511689	10·3196721	·0009099181
1100	1210000	1331000000	33·1662479	10·3228012	·0009090909
1101	1212201	1334633301	33·1813200	10·3259284	·0009082652
1102	1214404	1338273208	33·1963853	10·3290537	·0009074410
1103	1216609	1341919727	33·2114438	10·3321770	·0009066183
1104	1218816	1345572864	33·2264955	10·3352985	·0009057971
1105	1221025	1349232625	33·2415403	10·3384181	·0009049774
1106	1223236	1352899016	33·2565783	10·3415358	·0009041591
1107	1225449	1356572043	33·2716095	10·3446517	·0009033424
1108	1227664	1360251712	33·2866339	10·3477657	·0009025271
1109	1229881	1363938029	33·3016516	10·3508778	·0009017133
1110	1232100	1367631000	33·3166625	10·3539880	·0009009009
1111	1234321	1371330631	33·3316666	10·3570964	·0009000900
1112	1236544	1375036928	33·3466640	10·3602029	·0008992806
1113	1238769	1378749897	33·3616546	10·3633076	·0008984726
1114	1240996	1382469544	33·3766385	10·3664103	·0008976661
1115	1243225	1386195875	33·3916157	10·3695113	·0008968610
1116	1245456	1389928896	33·4065862	10·3726103	·0008960573
1117	1247689	1393668613	33·4215499	10·3757076	·0008952551
1118	1249924	1397415032	33·4365070	10·3788030	·0008944544
1119	1252161	1401168159	33·4514573	10·3818965	·0008936550
1120	1254400	1404928000	33·4664011	10·3849882	·0008928571
1121	1256641	1408694561	33·4813381	10·3880781	·0008920607
1122	1258884	1412467848	33·4962684	10·3911661	·0008912656
1123	1261129	1416247867	33·5111921	10·3942523	·0008904720

No.	Square	Cube	Square Root	Cube Root	Reciprocal
1124	1263376	1420034624	33·5261092	10·3973366	·0008896797
1125	1265625	1423828125	33·5410196	10·4004192	·0008888889
1126	1267876	1427628376	33·5559234	10·4034999	·0008880995
1127	1270129	1431435383	33·5708206	10·4065787	·0008873114
1128	1272384	1435249152	33·5857112	10·4096557	·0008865248
1129	1274641	1439069689	33·6005952	10·4127310	·0008857396
1130	1276900	1442897000	33·6154726	10·4158044	·0008849558
1131	1279161	1446731091	33·6303434	10·4188760	·0008841733
1132	1281424	1450571968	33·6452077	10·4219458	·0008833922
1133	1283689	1454419637	33·6600653	10·4250138	·0008826125
1134	1285956	1458274104	33·6749165	10·4280800	·0008818342
1135	1288225	1462135375	33·6897610	10·4311443	·0008810573
1136	1290496	1466003456	33·7045991	10·4342069	·0008802817
1137	1292769	1469878353	33·7194306	10·4372677	·0008795075
1138	1295044	1473760072	33·7342556	10·4403267	·0008787346
1139	1297321	1477648619	33·7490741	10·4433839	·0008779631
1140	1299600	1481544000	33·7638860	10·4464393	·0008771930
1141	1301881	1485446221	33·7786915	10·4494929	·0008764242
1142	1304164	1489355288	33·7934905	10·4525448	·0008756567
1143	1306449	1493271207	33·8082830	10·4555948	·0008748906
1144	1308736	1497193984	33·8230691	10·4586431	·0008741259
1145	1311025	1501123625	33·8378486	10·4616896	·0008733624
1146	1313316	1505060136	33·8526218	10·4647343	·0008726003
1147	1315609	1509003523	33·8673884	10·4677773	·0008718396
1148	1317904	1512953792	33·8821487	10·4708185	·0008710801
1149	1320201	1516910949	33·8969025	10·4738579	·0008703220
1150	1322500	1520875000	33·9116499	10·4768955	·0008695652
1151	1324801	1524845951	33·9263909	10·4799314	·0008688097
1152	1327104	1528823808	33·9411255	10·4829656	·0008680556
1153	1329409	1532808577	33·9558537	10·4859980	·0008673027
1154	1331716	1536800264	33·9705755	10·4890286	·0008665511
1155	1334025	1540798875	33·9852910	10·4920575	·0008658009
1156	1336336	1544804416	34·0000000	10·4950847	·0008650519
1157	1338649	1548816893	34·0147027	10·4981101	·0008643042
1158	1340964	1552836312	34·0293990	10·5011337	·0008635579
1159	1343281	1556862679	34·0440890	10·5041556	·0008628128
1160	1345600	1560896000	34·0587727	10·5071757	·0008620690
1161	1347921	1564936281	34·0734501	10·5101942	·0008613264
1162	1350244	1568983528	34·0881211	10·5132109	·0008605852
1163	1352569	1573037747	34·1027858	10·5162259	·0008598452
1164	1354896	1577098944	34·1174442	10·5192391	·0008591065
1165	1357225	1581167125	34·1320963	10·5222506	·0008583691
1166	1359556	1585242296	34·1467422	10·5252604	·0008576329
1167	1361889	1589324463	34·1613817	10·5282685	·0008568980
1168	1364224	1593413632	34·1760150	10·5312749	·0008561644
1169	1366561	1597509809	34·1906420	10·5342795	·0008554320
1170	1368900	1601613000	34·2052627	10·5372825	·0008547009
1171	1371241	1605723211	34·2198773	10·5402837	·0008539710
1172	1373584	1609840448	34·2344855	10·5432832	·0008532423

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No.	Square	Cube	Square Root	Cube Root	Reciprocal
1173	1375929	1613964717	34·2490875	10·5462810	·0008525149
1174	1378276	1618096024	34·2636834	10·5492771	·0008517888
1175	1380625	1622234375	34·2782730	10·5522715	·0008510638
1176	1382976	1626379776	34·2928564	10·5552642	·0008503401
1177	1385329	1630532233	34·3074336	10·5582552	·0008496177
1178	1387684	1634691752	34·3220046	10·5612445	·0008488964
1179	1390041	1638858339	34·3365694	10·5642322	·0008481764
1180	1392400	1643032000	34·3511281	10·5672181	·0008474576
1181	1394761	1647212741	34·3656805	10·5702024	·0008467401
1182	1397124	1651400568	34·3802268	10·5731849	·0008460237
1183	1399489	1655595487	34·3947670	10·5761658	·0008453085
1184	1401856	1659797504	34·4093011	10·5791449	·0008445946
1185	1404225	1664006625	34·4238289	10·5821225	·0008438819
1186	1406596	1668222856	34·4383507	10·5850983	·0008431703
1187	1408969	1672446203	34·4528663	10·5880725	·0008424600
1188	1411344	1676676672	34·4673759	10·5910450	·0008417508
1189	1413721	1680914269	34·4818793	10·5940158	·0008410429
1190	1416100	1685159000	34·4963766	10·5969850	·0008403361
1191	1418481	1689410871	34·5108678	10·5999525	·0008396306
1192	1420864	1693669888	34·5253530	10·6029184	·0008389262
1193	1423249	1697936057	34·5398321	10·6058826	·0008382230
1194	1425636	1702209384	34·5543051	10·6088451	·0008375209
1195	1428025	1706489875	34·5687720	10·6118060	·0008368201
1196	1430416	1710777536	34·5832329	10·6147652	·0008361204
1197	1432809	1715072373	34·5976879	10·6177228	·0008354219
1198	1435204	1719374392	34·6121366	10·6206788	·0008347245
1199	1437601	1723683599	34·6265794	10·6236331	·0008340284
1200	1440000	1728000000	34·6410162	10·6265857	·0008333333
1201	1442401	1732323601	34·6554469	10·6295367	·0008326395
1202	1444804	1736654408	34·6698716	10·6324860	·0008319468
1203	1447209	1740992427	34·6842904	10·6354338	·0008312552
1204	1449616	1745337664	34·6987031	10·6383799	·0008305648
1205	1452025	1749690125	34·7131099	10·6413244	·0008298755
1206	1454436	1754049816	34·7275107	10·6442672	·0008291874
1207	1456849	1758416743	34·7419055	10·6472085	·0008285004
1208	1459264	1762790912	34·7562944	10·6501480	·0008278146
1209	1461681	1767172329	34·7706773	10·6530860	·0008271299
1210	1464100	1771561000	34·7850543	10·6560223	·0008264463
1211	1466521	1775956931	34·7994253	10·6589570	·0008257638
1212	1468944	1780360128	34·8137904	10·6618902	·0008250825
1213	1471369	1784770597	34·8281495	10·6648217	·0008244023
1214	1473796	1789188344	34·8425028	10·6677516	·0008237232
1215	1476225	1793613375	34·8568501	10·6706799	·0008230453
1216	1478656	1798045696	34·8711915	10·6736066	·0008223684
1217	1481089	1802485313	34·8855271	10·6765317	·0008216927
1218	1483524	1806932232	34·8998567	10·6794552	·0008210181
1219	1485961	1811386459	34·9141805	10·6823771	·0008203445
1220	1488400	1815848000	34·9284984	10·6852973	·0008196721
1221	1490841	1820316861	34·9428104	10·6882160	·0008190008

No.	Square	Cube	Square Root	Cube Root	Reciprocal
1222	1493284	1824793048	34·9571166	10·6911331	·0008183306
1223	1495729	1829276567	34·9714169	10·6940486	·0008176615
1224	1498176	1833767424	34·9857114	10·6969625	·0008169935
1225	1500625	1838265625	35·0000000	10·6998748	·0008163265
1226	1503076	1842771176	35·0142828	10·7027855	·0008156607
1227	1505529	1847284083	35·0285598	10·7056947	·0008149959
1228	1507984	1851804352	35·0428309	10·7086023	·0008143322
1229	1510441	1856331989	35·0570963	10·7115083	·0008136696
1230	1512900	1860867000	35·0713558	10·7144127	·0008130081
1231	1515361	1865409391	35·0856096	10·7173155	·0008123477
1232	1517824	1869959168	35·0998575	10·7202168	·0008116883
1233	1520289	1874516337	35·1140997	10·7231165	·0008110300
1234	1522756	1879080904	35·1283361	10·7260146	·0008103728
1235	1525225	1883652875	35·1425668	10·7289112	·0008097166
1236	1527696	1888232256	35·1567917	10·7318062	·0008090615
1237	1530169	1892819053	35·1710108	10·7346997	·0008084074
1238	1532644	1897413272	35·1852242	10·7375916	·0008077544
1239	1535121	1902014919	35·1994318	10·7404819	·0008071025
1240	1537600	1906624000	35·2136337	10·7433707	·0008064516
1241	1540081	1911240521	35·2278299	10·7462579	·0008058018
1242	1542564	1915864488	35·2420204	10·7491436	·0008051530
1243	1545049	1920495907	35·2562051	10·7520277	·0008045052
1244	1547536	1925134784	35·2703842	10·8549103	·0008038585
1245	1550025	1929781125	35·2845575	10·7577913	·0008032129
1246	1552516	1934434936	35·2987252	10·7606708	·0008025682
1247	1555009	1939096223	35·3128872	10·7635488	·0008019246
1248	1557504	1943764992	35·3270435	10·7664252	·0008012821
1249	1560001	1948441249	35·3411941	10·7693001	·0008006405
1250	1562500	1953125000	35·3553391	10·7721735	·0008000000
1251	1565001	1957816251	35·3694784	10·7750453	·0007993605
1252	1567504	1962515008	35·3836120	10·7779156	·0007987220
1253	1570009	1967221277	35·3977400	10·7807843	·0007980846
1254	1572516	1971935064	35·4118624	10·7836516	·0007974482
1255	1575025	1976656375	35·4259792	10·7865173	·0007968127
1256	1577536	1981385216	35·4400903	10·7893815	·0007961783
1257	1580049	1986121593	35·4541958	10·7922441	·0007955449
1258	1582564	1990865512	35·4682957	10·7951053	·0007949126
1259	1585081	1995616979	35·4823900	10·7979649	·0007942812
1260	1587600	2000376000	35·4964787	10·8008230	·0007936508
1261	1590121	2005142581	35·5105618	10·8036797	·0007930214
1262	1592644	2009916728	35·5246393	10·8065348	·0007923930
1263	1595169	2014698447	35·5387113	10·8093884	·0007917656
1264	1597696	2019487744	35·5527777	10·8122404	·0007911392
1265	1600225	2024284625	35·5668385	10·8150909	·0007905138
1266	1602756	2029089096	35·5808937	10·8179400	·0007898894
1267	1605289	2033901163	35·5949434	10·8207876	·0007892660
1268	1607824	2038720832	35·6089876	10·8236336	·0007886435
1269	1610361	2043548109	35·6230262	10·8264782	·0007880221
1270	1612900	2048383000	35·6370593	10·8293213	·0007874016

No.	Square	Cube	Square Root	Cube Root	Reciprocal
1271	1615441	2053225511	35·6510869	10·8321629	·0007867821
1272	1617984	2058075648	35·6651090	10·8350030	·0007861635
1273	1620529	2062933417	35·6791255	10·8378416	·0007855460
1274	1623076	2067798824	35·6931366	10·8406788	·0007849294
1275	1625625	2072671875	35·7071421	10·8435144	·0007843137
1276	1628176	2077552576	35·7211422	10·8463485	·0007836991
1277	1630729	2082440933	35·7351367	10·8491812	·0007830854
1278	1633284	2087336952	35·7491258	10·8520125	·0007824726
1279	1635841	2092240639	35·7631095	10·8548422	·0007818608
1280	1638400	2097152000	35·7770876	10·8576704	·0007812500
1281	1640961	2102071041	35·7910603	10·8604972	·0007806401
1282	1643524	2106997768	35·8050276	10·8633225	·0007800312
1283	1646089	2111932187	35·8189894	10·8661464	·0007794232
1284	1648656	2116874304	35·8329457	10·8689687	·0007788162
1285	1651225	2121824125	35·8468966	10·8717897	·0007782101
1286	1653796	2126781656	35·8608421	10·8746091	·0007776050
1287	1656369	2131746903	35·8747822	10·8774271	·0007770008
1288	1658944	2136719872	35·8887169	10·8802436	·0007763975
1289	1661521	2141700569	35·9026461	10·8830587	·0007757952
1290	1664100	2146689000	35·9165699	10·8858723	·0007751938
1291	1666681	2151685171	35·9304884	10·8886845	·0007745933
1292	1669264	2156689088	35·9444015	10·8914952	·0007739938
1293	1671849	2161700757	35·9583092	10·8943044	·0007733952
1294	1674436	2166720184	33·9722115	10·8971123	·0007727975
1295	1677025	2171747375	35·9861084	10·8999186	·0007722008
1296	1679616	2176782336	36·0000000	10·9027235	·0007716049
1297	1682209	2181825073	36·0138862	10·9055269	·0007710100
1298	1684804	2186875592	36·0277671	10·9083290	·0007704160
1299	1687401	2191933899	36·0416426	10·9111296	·0007698229
1300	1690000	2197000000	36·0555128	10·9139287	·0007692308
1301	1692601	2202073901	36·0693776	10·9167265	·0007686395
1302	1695204	2207155608	36·0832371	10·9195228	·0007680492
1303	1697809	2212245127	36·0970913	10·9223177	·0007674597
1304	1700416	2217342464	36·1109402	10·9251111	·0007668712
1305	1703025	2222447625	36·1247837	10·9279031	·0007662835
1306	1705636	2227560616	36·1386220	10·9306937	·0007656968
1307	1708249	2232681443	36·1524550	10·9334829	·0007651109
1308	1710864	2237810112	36·1662826	10·9362706	·0007645260
1309	1713481	2242946629	36·1801050	10·9390569	·0007639419
1310	1716100	2248091000	36·1939221	10·9418418	·0007633588
1311	1718721	2253243231	36·2077340	10·9446253	·0007627765
1312	1721344	2258403328	36·2215406	10·9474074	·0007621951
1313	1723969	2263571297	36·2353419	10·9501880	·0007616146
1314	1726596	2268747144	36·2491379	10·9529673	·0007610350
1315	1729225	2273930875	36·2629287	10·9557451	·0007604563
1316	1731856	2279122496	36·2767143	10·9585215	·0007598784
1317	1734489	2284322013	36·2904946	10·9612965	·0007593014
1318	1737124	2289529432	36·3042697	10·9640701	·0007587253
1319	1739761	2294744759	36·3180396	10·9668423	·0007581501

No.	Square	Cube	Square Root	Cube Root	Reciprocal
1320	1742400	2299968000	36·3318042	10·9696131	·0007575758
1321	1745041	2305199161	36·3455637	10·9723825	·0007570023
1322	1747684	2310438248	36·3593179	10·9751505	·0007564297
1323	1750329	2315685267	36·3730670	10·9779171	·0007558579
1324	1752976	2320940224	36·3868108	10·9806823	·0007552870
1325	1755625	2326203125	36·4005494	10·9834462	·0007547170
1326	1758276	2331473976	36·4142829	10·9862086	·0007541478
1327	1760929	2336752783	36·4280112	10·9889696	·0007535795
1328	1763584	2342039552	36·4417343	10·9917293	·0007530120
1329	1766241	2347334289	36·4554523	10·9944876	·0007524454
1330	1768900	2352637000	36·4691650	10·9972445	·0007518797
1331	1771561	2357947691	36·4828727	11·0000000	·0007513148
1332	1774224	2363266368	36·4965752	11·0027541	·0007507508
1333	1776889	2368593037	36·5102725	11·0055069	·0007501875
1334	1779556	2373927704	36·5239647	11·0082583	·0007496252
1335	1782225	2379270375	36·5376518	11·0110082	·0007490637
1336	1784896	2384621056	36·5513338	11·0137569	·0007485030
1337	1787569	2389979753	36·5650106	11·0165041	·0007479432
1338	1790244	2395346472	36·5786823	11·0192500	·0007473842
1339	1792921	2400721219	36·5923489	11·0219945	·0007468260
1340	1795600	2406104000	36·6060104	11·0247377	·0007462687
1341	1798281	2411494821	36·6196668	11·0274795	·0007457122
1342	1800964	2416893688	36·6333181	11·0302199	·0007451565
1343	1803649	2422300607	36·6469644	11·0329590	·0007446016
1344	1806336	2427715584	36·6606056	11·0356967	·0007440476
1345	1809025	2433138625	36·6742416	11·0384330	·0007434944
1346	1811716	2438569736	36·6878726	11·0411680	·0007429421
1347	1814409	2444008923	36·7014986	11·0439017	·0007423905
1348	1817104	2449456192	36·7151195	11·0466339	·0007418398
1349	1819801	2454911549	36·7287353	11·0493649	·0007412898
1350	1822500	2460375000	36·7423461	11·0520945	·0007407407
1351	1825201	2465846551	36·7559519	11·0548227	·0007401924
1352	1827904	2471326208	36·7695526	11·0575497	·0007396450
1353	1830609	2476813977	36·7831483	11·0602752	·0007390983
1354	1833316	2482309864	36·7967390	11·0629994	·0007385524
1355	1836025	2487813875	36·8103246	11·0657222	·0007380074
1356	1838736	2493326016	36·8239053	11·0684437	·0007374631
1357	1841449	2498846293	36·8374809	11·0711639	·0007369197
1358	1844164	2504374712	36·8510515	11·0738828	·0007363770
1359	1846881	2509911279	36·8646172	11·0766003	·0007358352
1360	1849600	2515456000	36·8781778	11·0793165	·0007352941
1361	1852321	2521008881	36·8917335	11·0820314	·0007347539
1362	1855044	2526569928	36·9052842	11·0847449	·0007342144
1363	1857769	2532139147	36·9188299	11·0874571	·0007336757
1364	1860496	2537716544	36·9323706	11·0901679	·0007331378
1365	1863225	2543302125	36·9459064	11·0928775	·0007326007
1366	1865956	2548895896	36·9594372	11·0955857	·0007320644
1367	1868689	2554497863	36·9729631	11·0982926	·0007315289
1368	1871424	2560108032	36·9864840	11·1009982	·0007309942

No.	Square	Cube	Square Root	Cube Root	Reciprocal
1369	1874161	2565726409	37·0000000	11·1037025	·0007304602
1370	1876900	2571353000	37·0135110	11·1064054	·0007299270
1371	1879641	2576987811	37·0270172	11·1091070	·0007293946
1372	1882384	2582630848	37·0405184	11·1118073	·0007288630
1373	1885129	2588282117	37·0540146	11·1145064	·0007283321
1374	1887876	2593941624	37·0675060	11·1172041	·0007278020
1375	1890625	2599609375	37·0809924	11·1199004	·0007272727
1376	1893376	2605285376	37·0944740	11·1225955	·0007267442
1377	1896129	2610969633	37·1079506	11·1252893	·0007262164
1378	1898884	2616662152	37·1214224	11·1279817	·0007256894
1379	1901641	2622362939	37·1348893	11·1306729	·0007251632
1380	1904400	2628072000	37·1483512	11·1333628	·0007246377
1381	1907161	2633789341	37·1618084	11·1360514	·0007241130
1382	1909924	2639514968	37·1752606	11·1387386	·0007235890
1383	1912689	2645248887	37·1887079	11·1414246	·0007230658
1384	1915456	2650991104	37·2021505	11·1441093	·0007225434
1385	1918225	2656741625	37·2155881	11·1467926	·0007220217
1386	1920996	2662500456	37·2290209	11·1494747	·0007215007
1387	1923769	2668267603	37·2424489	11·1521555	·0007209805
1388	1926544	2674043072	37·2558720	11·1548350	·0007204611
1389	1929321	2679826869	37·2692903	11·1575133	·0007199424
1390	1932100	2685619000	37·2827037	11·1601903	·0007194245
1391	1934881	2691419471	37·2961124	11·1628659	·0007189073
1392	1937664	2697228288	37·3095162	11·1655403	·0007183908
1393	1940449	2703045457	37·3229152	11·1682134	·0007178751
1394	1943236	2708870984	37·3363094	11·1708852	·0007173601
1395	1946025	2714704875	37·3496988	11·1735558	·0007168459
1396	1948816	2720547136	37·3630834	11·1762250	·0007163324
1397	1951609	2726397773	37·3764632	11·1788930	·0007158196
1398	1954404	2732256792	37·3898382	11·1815598	·0007153076
1399	1957201	2738124199	37·4032084	11·1842252	·0007147963
1400	1960000	2744000000	37·4165738	11·1868894	·0007142857
1401	1962801	2749884201	37·4299345	11·1895523	·0007137759
1402	1965604	2755776808	37·4432904	11·1922139	·0007132668
1403	1968409	2761677827	37·4566416	11·1948743	·0007127584
1404	1971216	2767587264	37·4699880	11·1975334	·0007122507
1405	1974025	2773505125	37·4833296	11·2001913	·0007117438
1406	1976836	2779431416	37·4966665	11·2028479	·0007112376
1407	1979649	2785366143	37·5099987	11·2055032	·0007107321
1408	1982464	2791309312	37·5233261	11·2081573	·0007102273
1409	1985281	2797260929	37·5366487	11·2108101	·0007097232
1410	1988100	2803221000	37·5499667	11·2134617	·0007092199
1411	1990921	2809189531	37·5632799	11·2161120	·0007087172
1412	1993744	2815166528	37·5765885	11·2187611	·0007082153
1413	1996569	2821151997	37·5898922	11·2214089	·0007077141
1414	1999396	2827145944	37·6031913	11·2240554	·0007072136
1415	2002225	2833148375	37·6164857	11·2267007	·0007067138
1416	2005056	2839159296	37·6297754	11·2293448	·0007062147
1417	2007889	2845178713	37·6430604	11·2319876	·0007057163

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No.	Square	Cube	Square Root	Cube Root	Reciprocal
1418	2010724	2851206632	37·6563407	11·2346292	·0007052186
1419	2013561	2857243059	37·6696164	11·2372696	·0007047216
1420	2016400	2863288000	37·6828874	11·2399087	·0007042254
1421	2019241	2869341461	37·6961536	11·2425465	·0007037298
1422	2022084	2875403448	37·7094153	11·2451831	·0007032349
1423	2024929	2881473967	37·7226722	11·2478185	·0007027407
1424	2027776	2887553024	37·7359245	11·2504527	·0007022472
1425	2030625	2893640625	37·7491722	11·2530856	·0007017544
1426	2033476	2899736776	37·7624152	11·2557173	·0007012623
1427	2036329	2905841483	37·7756535	11·2583478	·0007007708
1428	2039184	2911954752	37·7888873	11·2609770	·0007002801
1429	2042041	2918076589	37·8021163	11·2636050	·0006997901
1430	2044900	2924207000	37·8153408	11·2662318	·0006993007
1431	2047761	2930345991	37·8285606	11·2688573	·0006988120
1432	2050624	2936493568	37·8417759	11·2714816	·0006983240
1433	2053489	2942649737	37·8549864	11·2741047	·0006978367
1434	2056356	2948814504	37·8681924	11·2767266	·0006973501
1435	2059225	2954987875	37·8813938	11·2793472	·0006968641
1436	2062096	2961169856	37·8945906	11·2819666	·0006963788
1437	2064969	2967360453	37·9077828	11·2845849	·0006958942
1438	2067844	2973559672	37·9209704	11·2872019	·0006954103
1439	2070721	2979767519	37·9341535	11·2898177	·0006949270
1440	2073600	2985984000	37·9473319	11·2924323	·0006944444
1441	2076481	2992209121	37·9605058	11·2950457	·0006939625
1442	2079364	2998442888	37·9736751	11·2976579	·0006934813
1443	2082249	3004685307	37·9868398	11·3002688	·0006930007
1444	2085136	3010936384	38·0000000	11·3028786	·0006925208
1445	2088025	3017196125	38·0131556	11·3054871	·0006920415
1446	2090916	3023464536	38·0263067	11·3080945	·0006915629
1447	2093809	3029741623	38·0394532	11·3107006	·0006910850
1448	2096704	3036027392	38·0525952	11·3133056	·0006906078
1449	2099601	3042321849	38·0657326	11·3159094	·0006901312
1450	2102500	3048625000	38·0788655	11·3185119	·0006896552
1451	2105401	3054936851	38·0919939	11·3211132	·0006891798
1452	2108304	3061257408	38·1051178	11·3237134	·0006887052
1453	2111209	3067586677	38·1182371	11·3263124	·0006882312
1454	2114116	3073924664	38·1313519	11·3289102	·0006877579
1455	2117025	3080271375	38·1444622	11·3315067	·0006872852
1456	2119936	3086626816	38·1575681	11·3341022	·0006868132
1457	2122849	3092990993	38·1706693	11·3366964	·0006863418
1458	2125764	3099363912	38·1837662	11·3392894	·0006858711
1459	2128681	3105745579	38·1968585	11·3418813	·0006854010
1460	2131600	3112136000	38·2099463	11·3444719	·0006849315
1461	2134521	3118535181	38·2230297	11·3470614	·0006844627
1462	2137444	3124943128	38·2361085	11·3496497	·0006839945
1463	2140369	3131359847	38·2491829	11·3522368	·0006835270
1464	2143296	3137785344	38·2622529	11·3548227	·0006830601
1465	2146225	3144219625	38·2753184	11·3574075	·0006825939
1466	2149156	3150662696	38·2883794	11·3599911	·0006821282

No.	Square	Cube	Square Root	Cube Root	Reciprocal
1467	2152089	3157114563	38·3014360	11·3625785	·0006816683
1468	2155024	3163575232	38·3144881	11·3651547	·0006811989
1469	2157961	3170044709	38·3275358	11·3677347	·0006807352
1470	2160900	3176523000	38·3405790	11·3703136	·0006802721
1471	2163841	3183010111	38·3536178	11·3728914	·0006798097
1472	2166784	3189506048	38·3666522	11·3754679	·0006793478
1473	2169729	3196010817	38·3796821	11·3780433	·0006788866
1474	2172676	3202524424	38·3927076	11·3806175	·0006784261
1475	2175625	3209046875	38·4057287	11·3831906	·0006779661
1476	2178576	3215578176	38·4187454	11·3857625	·0006775068
1477	2181529	3222118333	38·4317577	11·3883332	·0006770481
1478	2184484	3228667352	38·4447656	11·3909028	·0006765900
1479	2187441	3235225239	38·4577691	11·3934712	·0006761325
1480	2190400	3241792000	38·4707681	11·3960384	·0006756757
1481	2193361	3248367641	38·4837627	11·3986045	·0006752194
1482	2196324	3254952168	38·4967530	11·4011695	·0006747638
1483	2199289	3261545587	38·5097390	11·4037332	·0006743088
1484	2202256	3268147904	38·5227206	11·4062959	·0006738544
1485	2205225	3274759125	38·5356977	11·4088574	·0006734007
1486	2208196	3281379256	38·5486705	11·4114177	·0006729475
1487	2211169	3288008303	38·5616389	11·4139769	·0006724950
1488	2214144	3294646272	38·5746030	11·4165349	·0006720430
1489	2217121	3301293169	38·5875627	11·4190918	·0006715917
490	2220100	3307949000	38·6005181	11·4216476	·0006711409
491	2223081	3314613771	38·6134691	11·4242022	·0006706908
492	2226064	3321287488	38·6264158	11·4267556	·0006702413
493	2229049	3327970157	38·6393582	11·4293079	·0006697924
494	2232036	3334661784	38·6522962	11·4318591	·0006693440
495	2235025	3341362375	38·6652299	11·4344092	·0006688963
96	2238016	3348071936	38·6781593	11·4369581	·0006684492
97	2241009	3354790473	38·6910843	11·4395059	·0006680027
98	2244004	3361517992	38·7040050	11·4420525	·0006675567
99	2247001	3368254499	38·7169214	11·4445980	·0006671114
00	2250000	3375000000	38·7298335	11·4471424	·0006666667
01	2253001	3381754501	38·7427412	11·4496857	·0006662225
2	2256004	3388518008	38·7556447	11·4522278	·0006657790
3	2259009	3395290527	38·7685439	11·4547688	·0006643360
4	2262016	3402072064	38·7814389	11·4573087	·0006648936
5	2265025	3408862625	38·7943294	11·4598474	·0006644518
	2268036	3415662216	38·8072158	11·4623850	·0006640106
	2271049	3422470843	38·8200978	11·4649215	·0006635700
	2274064	3429288512	38·8329757	11·4674568	·0006631300
	2277081	3436115229	38·8458491	11·4699911	·0006626905
	2280100	3442951000	38·8587184	11·4725242	·0006622517
	2283121	3449795831	38·8715834	11·4750562	·0006618134
	2286144	3456649728	38·8844442	11·4775871	·0006613757
	2289169	3463512697	38·8973006	11·4801169	·0006609385
	2292196	3470384744	38·9101529	11·4826455	·0006605020
	2295225	3477265875	38·9230009	11·4851731	·0006600660

No.	Square	Cube	Square Root	Cube Root	Reciprocal
1516	2298256	3484156096	38·9358447	11·4876995	·0006596306
1517	2301289	3491055413	38·9486841	11·4902249	·0006591958
1518	2304324	3497963832	38·9615194	11·4927491	·0006587615
1519	2307361	3504881359	38·9743505	11·4952722	·0006583278
1520	2310400	3511808000	38·9871774	11·4977942	·0006578947
1521	2313441	3518743761	39·0000000	11·5003151	·0006574622
1522	2316484	3525688648	39·0128184	11·5028348	·0006570302
1523	2319529	3532642667	39·0256326	11·5053535	·0006565988
1524	2322576	3539605824	39·0384426	11·5078711	·0006561680
1525	2325625	3546578125	39·0512483	11·5103876	·0006557377
1526	2328676	3553559576	39·0640499	11·5129030	·0006553080
1527	2331729	3560550183	39·0768473	11·5154173	·0006548788
1528	2334784	3567549952	39·0896406	11·5179305	·0006544503
1529	2337841	3574558889	39·1024296	11·5204425	·0006540222
1530	2340900	3581577000	39·1152144	11·5229535	·0006535948
1531	2343961	3588604291	39·1279951	11·5254634	·0006531679
1532	2347024	3595640768	39·1407716	11·5279722	·0006527415
1533	2350089	3602686437	39·1535439	11·5304799	·0006523157
1534	2353156	3609741304	39·1663120	11·5329865	·0006518905
1535	2356225	3616805375	39·1790760	11·5354920	·0006514658
1536	2359296	3623878656	39·1918359	11·5379965	·0006510417
1537	2362369	3630961153	39·2045915	11·5404998	·0006506181
1538	2365444	3638052872	39·2173431	11·5430021	·0006501951
1539	2368521	3645153819	39·2300905	11·5455033	·0006497726
1540	2371600	3652264000	39·2428337	11·5480034	·0006493506
1541	2374681	3659383421	39·2555728	11·5505025	·0006489293
1542	2377764	3666512088	39·2683078	11·5530004	·0006485084
1543	2380849	3673650007	39·2810387	11·5554973	·0006480881
1544	2383936	3680797184	39·2937654	11·5579931	·0006476684
1545	2387025	3687953625	39·3064880	11·5604878	·0006472492
1546	2390116	3695119336	39·3192065	11·5629815	·0006468305
1547	2393209	3702294323	39·3319208	11·5654740	·0006464124
1548	2396304	3709478592	39·3446311	11·5679655	·0006459948
1549	2399401	3716672149	39·3573373	11·5704559	·0006455778
1550	2402500	3723875000	39·3700394	11·5729453	·0006451613
1551	2405601	3731087151	39·3827373	11·5754336	·0006447453
1552	2408704	3738308608	39·3954312	11·5779208	·0006443299
1553	2411809	3745539377	39·4081210	11·5804069	·0006439150
1554	2414916	3752779464	39·4208067	11·5828919	·0006435006
1555	2418025	3760028875	39·4334883	11·5853759	·0006430868
1556	2421136	3767287616	39·4461658	11·5878588	·0006426735
1557	2424249	3774555693	39·4588393	11·5903407	·0006422608
1558	2427364	3781833112	39·4715087	11·5928215	·0006418485
1559	2430481	3789119879	39·4841740	11·5953013	·0006414368
1560	2433600	3796416000	39·4968353	11·5977799	·0006410256
1561	2436721	3803721481	39·5094925	11·6002576	·0006406150
1562	2439844	3811036328	39·5221457	11·6027342	·0006402049
1563	2442969	3818360547	39·5347948	11·6052097	·0006397953
1564	2446096	3825694144	39·5474399	11·6076841	·0006393862

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No.	Square	Cube	Square Root	Cube Root	Reciprocal
1565	2449225	3833037125	39·5600809	11·6101575	·0006389776
1566	2452356	3840389496	39·5727179	11·6126299	·0006385696
1567	2455489	3847751263	39·5853508	11·6151012	·0006381621
1568	2458624	3855122432	39·5979797	11·6175715	·0006377551
1569	2461761	3862503009	39·6106046	11·6200407	·0006373486
1570	2464900	3869893000	39·6232255	11·6225088	·0006369427
1571	2468041	3877292411	39·6358424	11·6249759	·0006365372
1572	2471184	3884701248	39·6484552	11·6274420	·0006361323
1573	2474329	3892119517	39·6610640	11·6299070	·0006357279
1574	2477476	3899547224	39·6736688	11·6323710	·0006353240
1575	2480625	3906984375	39·6862696	11·6348339	·0006349206
1576	2483776	3914430976	39·6988665	11·6372957	·0006345178
1577	2486929	3921887033	39·7114593	11·6397566	·0006341154
1578	2490084	3929352552	39·7240481	11·6422164	·0006337136
1579	2493241	3936827539	39·7366329	11·6446751	·0006333122
1580	2496400	3944312000	39·7492138	11·6471329	·0006329114
1581	2499561	3951805941	39·7617907	11·6495895	·0006325111
1582	2502724	3959309368	39·7743636	11·6520452	·0006321113
1583	2505889	3966822287	39·7869325	11·6544998	·0006317119
1584	2509056	3974344704	39·7994975	11·6569534	·0006343131
1585	2512225	3981876625	39·8120585	11·6594059	·0006309148
1586	2515396	3989418056	39·8246155	11·6618574	·0006305170
1587	2518569	3996969003	39·8371686	11·6643079	·0006301197
1588	2521744	4004529472	39·8497177	11·6667574	·0006297229
1589	2524921	4012099469	39·8622628	11·6692058	·0006293266
1590	2528100	4019679000	39·8748040	11·6716532	·0006289308
1591	2531281	4027268071	39·8873413	11·6740996	·0006285355
1592	2534464	4034866688	39·8998747	11·6765449	·0006281407
1593	2537649	4042474857	39·9124041	11·6789892	·0006277464
1594	2540836	4050092584	39·9249295	11·6814325	·0006273526
1595	2544025	4057719875	39·9374511	11·6838748	·0006269592
1596	2547216	4065356736	39·9499687	11·6863161	·0006265664
1597	2550409	4073003173	39·9624824	11·6887563	·0006261741
1598	2553604	4080659192	39·9749922	11·6911955	·0006257822
1599	2556801	4088324799	39·9874980	11·6936337	·0006253909
1600	2560000	4096000000	40·0000000	11·6960709	·0006250000
1601	2563201	4103684801	40·0124980	11·6985071	·0006246096
1602	2566404	4111379208	40·0249922	11·7009422	·0006242197
1603	2569609	4119083227	40·0374824	11·7033764	·0006238303
1604	2572816	4126796864	40·0499688	11·7058095	·0006234414
1605	2576025	4134520125	40·0624512	11·7082417	·0006230530
1606	2579236	4142253016	40·0749298	11·7106728	·0006226650
1607	2582449	4149995543	40·0874045	11·7131029	·0006222775
1608	2585664	4157747712	40·0998753	11·7155320	·0006218905
1609	2588881	4165509529	40·1123423	11·7179601	·0006215040
1610	2592100	4173281000	40·1248053	11·7203872	·0006211180
1611	2595321	4181062131	40·1372645	11·7228133	·0006207325
1612	2598544	4188852928	40·1497198	11·7252384	·0006203474
1613	2601769	4196653397	40·1621713	11·7276625	·0006199628

No.	Square	Cube	Square Root	Cube Root	Reciprocal
1614	2604996	4204463544	40·1746188	11·7300855	·0006195787
1615	2608225	4212283375	40·1870626	11·7325076	·0006191950
1616	2611456	4220112896	40·1995025	11·7349286	·0006188119
1617	2614689	4227952113	40·2119385	11·7373487	·0006184292
1618	2617924	4235801032	40·2243707	11·7397677	·0006180470
1619	2621161	4243659659	40·2367990	11·7421858	·0006176652
1620	2624400	4251528000	40·2492236	11·7446029	·0006172840
1621	2627641	4259406061	40·2616443	11·7470190	·0006169031
1622	2630884	4267293848	40·2740611	11·7494341	·0006165228
1623	2634129	4275191367	40·2864742	11·7518482	·0006161429
1624	2637376	4283098624	40·2988834	11·7542613	·0006157635
1625	2640625	4291015625	40·3112888	11·7566734	·0006153846
1626	2643876	4298942376	40·3236903	11·7590846	·0006150062
1627	2647129	4306878883	40·3360881	11·7614947	·0006146282
1628	2650384	4314825152	40·3484820	11·7639039	·0006142506
1629	2653641	4322781189	40·3608721	11·7663121	·0006138735
1630	2656900	4330747000	40·3732585	11·7687193	·0006134969
1631	2660161	4338722591	40·3856410	11·7711255	·0006131208
1632	2663424	4346707968	40·3980198	11·7735306	·0006127451
1633	2666689	4354703137	40·4103947	11·7759349	·0006123699
1634	2669956	4362708104	40·4227658	11·7783381	·0006119951
1635	2673225	4370722875	40·4351332	11·7807404	·0006116208
1636	2676496	4378747456	40·4474968	11·7831417	·0006112469
1637	2679769	4386781853	40·4598566	11·7855420	·0006108735
1638	2683044	4394826072	40·4722127	11·7879414	·0006105006
1639	2686321	4402880119	40·4845649	11·7903397	·0006101281
1640	2689600	4410944000	40·4969135	11·7927371	·0006097561
1641	2692881	4419017721	40·5092582	11·7951335	·0006093845
1642	2696164	4427101288	40·5215992	11·7975289	·0006090134
1643	2699449	4435194707	40·5339364	11·7999234	·0006086427
1644	2702736	4443297984	40·5462699	11·8023169	·0006082725
1645	2706025	4451411125	40·5585996	11·8047094	·0006079027
1646	2709316	4459534136	40·5709255	11·8071010	·0006075334
1647	2712609	4467667023	40·5832477	11·8094916	·0006071645
1648	2715904	4475809792	40·5955663	11·8118812	·0006067961
1649	2719201	4483962449	40·6078810	11·8142698	·0006064281
1650	2722500	4492125000	40·6201920	11·8166576	·0006060606
1651	2725801	4500297451	40·6324993	11·8190443	·0006056935
1652	2729104	4508479808	40·6448029	11·8214301	·0006053269
1653	2732409	4516672077	40·6571027	11·8238149	·0006049607
1654	2735716	4524874264	40·6693988	11·8261987	·0006045949
1655	2739025	4533086375	40·6816912	11·8285816	·0006042296
1656	2742336	4541308416	40·6939799	11·8309634	·0006038647
1657	2745649	4549540393	40·7062648	11·8333444	·0006035003
1658	2748964	4557782312	40·7185461	11·8357244	·0006031363
1659	2752281	4566034179	40·7308237	11·8381034	·0006027728
1660	2755600	4574296000	40·7430976	11·8404815	·0006024096
1661	2758921	4582567781	40·7553677	11·8428586	·0006020470
1662	2762244	4590849528	40·7676342	11·8452348	·0006016847

No.	Square	Cube	Square Root	Cube Root	Reciprocal
1663	2765569	4599141247	40·7798970	11·8476100	·0006013229
1664	2768896	4607442994	40·7921561	11·8499843	·0006009615
1665	2772225	4615754625	40·8044115	11·8523576	·0006006006
1666	2775556	4624076296	40·8166633	11·8547299	·0006002401
1667	2778889	4632407963	40·8289113	11·8571014	·0005998800
1668	2782224	4640749632	40·8411557	11·8594719	·0005995204
1669	2785561	4649101309	40·8533964	11·8618414	·0005991612
1670	2788900	4657463000	40·8656335	11·8642100	·0005988024
1671	2792241	4665834711	40·8778669	11·8665776	·0005984440
1672	2795584	4674216448	40·8900966	11·8689443	·0005980861
1673	2798929	4682608217	40·9023227	11·8713100	·0005977286
1674	2802276	4691010024	40·9145451	11·8736748	·0005973716
1675	2805625	4699421875	40·9267638	11·8760387	·0005970149
1676	2808976	4707843776	40·9389790	11·8784016	·0005966587
1677	2812329	4716275733	40·9511905	11·8807636	·0005963029
1678	2815684	4724717752	40·9633983	11·8831246	·0005959476
1679	2819041	4733169839	40·9756025	11·8854847	·0005955926
1680	2822400	4741632000	40·9878031	11·8878439	·0005952381
1681	2825761	4750104241	41·0000000	11·8902022	·0005948840
1682	2829124	4758586568	41·0121933	11·8925595	·0005945303
1683	2832489	4767078987	41·0243830	11·8949159	·0005941771
1684	2835856	4775581504	41·0365691	11·8972713	·0005938242
1685	2839225	4784094125	41·0487515	11·8996258	·0005934718
1686	2842596	4792616856	41·0609303	11·9019793	·0005931198
1687	2845969	4801149703	41·0731055	11·9043319	·0005927682
1688	2849344	4809692672	41·0852772	11·9066836	·0005924171
1689	2852721	4818245769	41·0974452	11·9090344	·0005920663
1690	2856100	4826809000	41·1096096	11·9113843	·0005917160
1691	2859481	4835382371	41·1217704	11·9137332	·0005913661
1692	2862864	4843965888	41·1339276	11·9160812	·0005910165
1693	2866249	4852559557	41·1460812	11·9184283	·0005906675
1694	2869636	4861163384	41·1582313	11·9207744	·0005903188
1695	2873025	4869777375	41·1703777	11·9231196	·0005899705
1696	2876416	4878401536	41·1825206	11·9254639	·0005896226
1697	2879809	4887035873	41·1946599	11·9278073	·0005892752
1698	2883204	4895680392	41·2067956	11·9301497	·0005889282
1699	2886601	4904335099	41·2189277	11·9324913	·0005885815
1700	2890000	4913000000	41·2310563	11·9348319	·0005882353
1701	2893401	4921675101	41·2431812	11·9371716	·0005878895
1702	2896804	4930360408	41·2553027	11·9395104	·0005875441
1703	2900209	4939055927	41·2674205	11·9418482	·0005871991
1704	2903616	4947761664	41·2795349	11·9441852	·0005868545
1705	2907025	4956477625	41·2916456	11·9465213	·0005865103
1706	2910436	4965203816	41·3037529	11·9488564	·0005861665
1707	2913849	4973940243	41·3158565	11·9511906	·0005858231
1708	2917264	4982686912	41·3279566	11·9535239	·0005854801
1709	2920681	4991443829	41·3400532	11·9558563	·0005851375
1710	2924100	5000211000	41·3521463	11·9581878	·0005847953
1711	2927521	5008988431	41·3642358	11·9605184	·0005844535

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No.	Square	Cube	Square Root	Cube Root	Reciprocal
1712	2930944	5017776128	41·3763217	11·9628481	·0005841121
1713	2934369	5026574097	41·3884042	11·9651768	·0005837712
1714	2937796	5035382344	41·4004831	11·9675047	·0005834306
1715	2941225	5044200875	41·4125585	11·9698317	·0005830904
1716	2944656	5053029696	41·4246304	11·9721577	·0005827506
1717	2948089	5061868813	41·4366987	11·9744829	·0005824112
1718	2951524	5070718232	41·4487636	11·9768071	·0005820722
1719	2954961	5079577959	41·4608249	11·9791304	·0005817336
1720	2958400	5088448000	41·4728827	11·9814528	·0005813953
1721	2961841	5097328361	41·4849370	11·9837744	·0005810575
1722	2965284	5106219048	41·4969878	11·9860950	·0005807201
1723	2968729	5115120067	41·5090351	11·9884148	·0005803831
1724	2972176	5124031424	41·5210790	11·9907336	·0005800464
1725	2975625	5132953125	41·5331193	11·9930516	·0005797101
1726	2979076	5141885176	41·5451561	11·9953686	·0005793743
1727	2982529	5150827583	41·5571895	11·9976848	·0005790388
1728	2985984	5159780352	41·5692194	12·0000000	·0005787037
1729	2989441	5168743489	41·5812457	12·0023144	·0005783690
1730	2992900	5177717000	41·5932686	12·0046278	·0005780347
1731	2996361	5186700891	41·6052881	12·0069404	·0005777008
1732	2999824	5195695168	41·6173041	12·0092521	·0005773672
1733	3003289	5204699837	41·6293166	12·0115629	·0005770340
1734	3006756	5213714904	41·6413256	12·0138728	·0005767013
1735	3010225	5222740375	41·6533312	12·0161818	·0005763689
1736	3013696	5231776256	41·6653333	12·0184900	·0005760369
1737	3017169	5240822553	41·6773319	12·0207973	·0005757052
1738	3020644	5249879272	41·6893271	12·0231037	·0005753740
1739	3024121	5258946419	41·7013189	12·0254092	·0005750431
1740	3027600	5268024000	41·7133072	12·0277138	·0005747126
1741	3031081	5277112021	41·7252921	12·0300175	·0005743825
1742	3034564	5286210488	41·7372735	12·0323204	·0005740528
1743	3038049	5295319407	41·7492515	12·0346223	·0005737235
1744	3041536	5304438784	41·7612260	12·0369233	·0005733945
1745	3045025	5313568625	41·7731971	12·0392235	·0005730659
1746	3048516	5322708936	41·7851648	12·0415229	·0005727377
1747	3052009	5331859723	41·7971291	12·0438213	·0005724098
1748	3055504	5341020992	41·8090899	12·0461189	·0005720824
1749	3059001	5350192749	41·8210473	12·0484156	·0005717553
1750	3062500	5359375000	41·8330013	12·0507114	·0005714286
1751	3066001	5368567751	41·8449519	12·0530063	·0005711022
1752	3069504	5377771008	41·8568991	12·0553003	·0005707763
1753	3073009	5386984777	41·8688428	12·0575935	·0005704507
1754	3076516	5396209064	41·8807832	12·0598859	·0005701254
1755	3080025	5405443875	41·8927201	12·0621778	·0005698006
1756	3083536	5414689216	41·9046537	12·0644679	·0005694761
1757	3087049	5423945093	41·9165838	12·0667576	·0005691520
1758	3090564	5433211512	41·9285106	12·0690464	·0005688282
1759	3094081	5442488479	41·9404339	12·0713344	·0005685048
1760	3097600	5451776000	41·9523539	12·0736215	·0005681818

No.	Square	Cube	Square Root	Cube Root	Reciprocal
1761	3101121	5461074081	41·9642705	12·0759077	·0005678592
1762	3104644	5470382728	41·9761837	12·0781930	·0005675369
1763	3108169	5479701947	41·9880935	12·0804775	·0005672150
1764	3111696	5489031744	42·0000000	12·0827612	·0005668934
1765	3115225	5498372125	42·0119031	12·0850439	·0005665722
1766	3118756	5507723096	42·0238028	12·0873258	·0005662514
1767	3122289	5517084663	42·0356991	12·0896069	·0005659310
1768	3125824	5526456832	42·0475921	12·0918870	·0005656109
1769	3129361	5535839609	42·0594817	12·0941664	·0005652911
1770	3132900	5545233000	42·0713679	12·0964449	·0005649718
1771	3136441	5554637011	42·0832508	12·0987226	·0005646527
1772	3139984	5564051648	42·0951304	12·1009993	·0005643341
1773	3143529	5573476917	42·1070065	12·1032753	·0005640158
1774	3147076	5582912824	42·1188794	12·1055503	·0005636979
1775	3150625	5592359375	42·1307488	12·1078245	·0005633803
1776	3154176	5601816576	42·1426150	12·1100979	·0005630631
1777	3157729	5611284433	42·1544778	12·1123704	·0005627462
1778	3161284	5620762952	42·1663373	12·1146420	·0005624297
1779	3164841	5630252139	42·1781934	12·1169128	·0005621135
1780	3168400	5639752000	42·1900462	12·1191827	·0005617978
1781	3171961	5649262541	42·2018957	12·1214518	·0005614823
1782	3175524	5658783768	42·2137418	12·1237200	·0005611672
1783	3179089	5668315687	42·2255846	12·1259874	·0005608525
1784	3182656	5677858304	42·2374242	12·1282539	·0005605381
1785	3186225	5687411625	42·2492603	12·1305197	·0005602241
1786	3189796	5696975656	42·2610932	12·1327845	·0005599104
1787	3193369	5706550403	42·2729227	12·1350485	·0005595971
1788	3196944	5716135872	42·2847490	12·1373117	·0005592841
1789	3200521	5725732069	42·2965719	12·1395740	·0005589715
1790	3204100	5735339000	42·3083916	12·1418355	·0005586592
1791	3207681	5744956671	42·3202079	12·1440961	·0005583473
1792	3211264	5754585088	42·3320210	12·1463559	·0005580357
1793	3214849	5764224257	42·3438307	12·1486148	·0005577245
1794	3218436	5773874184	42·3556371	12·1508729	·0005574136
1795	3222025	5783534875	42·3674403	12·1531302	·0005571031
1796	3225616	5793206336	42·3792402	12·1553866	·0005567929
1797	3229209	5802888573	42·3910368	12·1576422	·0005564830
1798	3232804	5812581592	42·4028301	12·1598970	·0005561735
1799	3236401	5822285399	42·4146201	12·1621509	·0005558644
1800	3240000	5832000000	42·4264069	12·1644040	·0005555556
1801	3243601	5841725401	42·4381903	12·1666562	·0005552471
1802	3247204	5851461608	42·4499705	12·1689076	·0005549390
1803	3250809	5861208627	42·4617475	12·1711582	·0005546312
1804	3254416	5870966464	42·4735212	12·1734079	·0005543237
1805	3258025	5880735125	42·4852916	12·1756569	·0005540166
1806	3261636	5890514616	42·4970587	12·1779050	·0005537099
1807	3265249	5900304943	42·5088226	12·1801522	·0005534034
1808	3268864	5910106112	42·5205833	12·1823987	·0005530973
1809	3272481	5919918129	42·5323406	12·1846443	·0005527916

No.	Square	Cube	Square Root	Cube Root	Reciprocal
1810	3276100	5929741000	42·5440948	12·1868891	·0005524862
1811	3279721	5939574731	42·5558456	12·1891331	·0005521811
1812	3283344	5949419328	42·5675933	12·1913762	·0005518764
1813	3286969	5959274797	42·5793377	12·1936185	·0005515720
1814	3290596	5969141144	42·5910789	12·1958599	·0005512679
1815	3294225	5979018375	42·6028168	12·1981006	·0005509642
1816	3297856	5988906496	42·6145515	12·2003404	·0005506608
1817	3301489	5998805513	42·6262829	12·2025794	·0005503577
1818	3305124	6008715432	42·6380112	12·2048176	·0005500550
1819	3308761	6018636259	42·6497362	12·2070549	·0005497526
1820	3312400	6028568000	42·6614580	12·2092915	·0005494505
1821	3316041	6038510661	42·6731766	12·2115272	·0005491488
1822	3319684	6048464248	42·6848919	12·2137621	·0005488474
1823	3323329	6058428767	42·6966040	12·2159962	·0005485464
1824	3326976	6068404224	42·7083130	12·2182295	·0005482456
1825	3330625	6078390625	42·7200187	12·2204620	·0005479452
1826	3334276	6088387976	42·7317212	12·2226936	·0005476451
1827	3337929	6098396283	42·7434206	12·2249244	·0005473454
1828	3341584	6108415552	42·7551167	12·2271544	·0005470460
1829	3345241	6118445789	42·7668095	12·2293836	·0005467469
1830	3348900	6128487000	42·7784992	12·2316120	·0005464481
1831	3352561	6138539191	42·7901858	12·2338396	·0005461496
1832	3356224	6148602368	42·8018691	12·2360663	·0005458515
1833	3359889	6158676537	42·8135492	12·2382923	·0005455537
1834	3363556	6168761704	42·8252262	12·2405174	·0005452563
1835	3367225	6178857875	42·8368999	12·2427418	·0005449591
1836	3370896	6188965056	42·8485706	12·2449653	·0005446623
1837	3374569	6199083253	42·8602380	12·2471880	·0005443658
1838	3378244	6209212472	42·8719022	12·2494099	·0005440696
1839	3381921	6219352719	42·8835633	12·2516310	·0005437738
1840	3385600	6229504000	42·8952212	12·2538513	·0005434783
1841	3389281	6239666321	42·9068759	12·2560708	·0005431831
1842	3392964	6249839688	42·9185275	12·2582895	·0005428882
1843	3396649	6260024107	42·9301759	12·2605074	·0005425936
1844	3400336	6270219584	42·9418211	12·2627245	·0005422993
1845	3404025	6280426125	42·9534632	12·2649408	·0005420054
1846	3407716	6290643736	42·9651021	12·2671563	·0005417118
1847	3411409	6300872423	42·9767379	12·2693710	·0005414185
1848	3415104	6311112192	42·9883705	12·2715849	·0005411255
1849	3418801	6321363049	43·0000000	12·2737980	·0005408329
1850	3422500	6331625000	43·0116263	12·2760103	·0005405405
1851	3426201	6341898051	43·0232495	12·2782218	·0005402485
1852	3429904	6352182208	43·0348696	12·2804325	·0005399568
1853	3433609	6362477477	43·0464865	12·2826424	·0005396654
1854	3437316	6372783864	43·0581003	12·2848515	·0005393743
1855	3441025	6383101375	43·0697109	12·2870598	·0005390836
1856	3444736	6393430016	43·0813185	12·2892673	·0005387931
1857	3448449	6403769793	43·0929228	12·2914740	·0005385030
1858	3452164	6414120712	43·1045241	12·2936800	·0005382131

No.	Square	Cube	Square Root	Cube Root	Reciprocal
1859	3455881	6424482779	43·1161223	12·2958851	·0005379236
1860	3459600	6434856000	43·1277173	12·2980895	·0005376344
1861	3463321	6445240381	43·1393092	12·3002930	·0005373455
1862	3467044	6455635928	43·1508980	12·3024958	·0005370569
1863	3470769	6466042647	43·1624837	12·3046978	·0005367687
1864	3474496	6476460544	43·1740663	12·3068990	·0005364807
1865	3478225	6486889625	43·1856458	12·3090994	·0005361930
1866	3481956	6497329896	43·1972221	12·3112991	·0005359057
1867	3485689	6507781363	43·2087954	12·3134979	·0005356186
1868	3489424	6518244032	43·2203656	12·3156959	·0005353319
1869	3493161	6528717909	43·2319326	12·3178932	·0005350455
1870	3496900	6539203000	43·2434966	12·3200897	·0005347594
1871	3500641	6549699311	43·2550575	12·3222854	·0005344735
1872	3504384	6560206848	43·2666153	12·3244803	·0005341880
1873	3508129	6570725617	43·2781700	12·3266744	·0005339028
1874	3511876	6581255624	43·2897216	12·3288678	·0005336179
1875	3515625	6591796875	43·3012702	12·3310604	·0005333333
1876	3519376	6602349376	43·3128157	12·3332522	·0005330490
1877	3523129	6612913133	43·3243580	12·3354432	·0005327651
1878	3526884	6623488152	43·3358973	12·3376334	·0005324814
1879	3530641	6634074439	43·3474336	12·3398229	·0005321980
1880	3534400	6644672000	43·3589668	12·3420116	·0005319149
1881	3538161	6655280841	43·3704969	12·3441995	·0005316321
1882	3541924	6665900968	43·3820239	12·3463866	·0005313496
1883	3545689	6676532387	43·3935479	12·3485730	·0005310674
1884	3549456	6687175104	43·4050688	12·3507586	·0005307856
1885	3553225	6697829125	43·4165867	12·3529434	·0005305040
1886	3556996	6708494456	43·4281015	12·3551274	·0005302227
1887	3560769	6719171103	43·4396132	12·3573107	·0005299417
1888	3564544	6729859072	43·4511220	12·3594932	·0005296610
1889	3568321	6740558369	43·4626276	12·3616749	·0005293806
1890	3572100	6751269000	43·4741302	12·3638559	·0005291005
1891	3575881	6761990971	43·4856298	12·3660361	·0005288207
1892	3579664	6772724288	43·4971263	12·3682155	·0005285412
1893	3583449	6783468957	43·5086198	12·3703941	·0005282620
1894	3587236	6794224984	43·5201103	12·3725721	·0005279831
1895	3591025	6804992375	43·5315977	12·3747492	·0005277045
1896	3594816	6815771136	43·5430821	12·3769255	·0005274262
1897	3598609	6826561273	43·5545635	12·3791011	·0005271481
1898	3602404	6837362792	43·5660418	12·3812759	·0005268704
1899	3606201	6848175699	43·5775171	12·3834500	·0005265929
1900	3610000	6859000000	43·5889894	12·3856233	·0005263158
1901	3613801	6869835701	43·6004587	12·3877959	·0005260389
1902	3617604	6880682808	43·6119249	12·3899676	·0005257624
1903	3621409	6891541327	43·6233882	12·3921386	·0005254861
1904	3625216	6902411264	43·6348485	12·3943089	·0005252101
1905	3629025	6913292625	43·6463057	12·3964784	·0005249344
1906	3632836	6924185416	43·6577599	12·3986471	·0005246590
1907	3636649	6935089643	43·6692111	12·4008151	·0005243838

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No.	Square	Cube	Square Root	Cube Root	Reciprocal
1908	3640464	6946005312	43·6806593	12·4029823	·0005241090
1909	3644281	6956932429	43·6921045	12·4051488	·0005238345
1910	3648100	6967871000	43·7035467	12·4073145	·0005235602
1911	3651921	6978821031	43·7149860	12·4094794	·0005232862
1912	3655744	6989782528	43·7264222	12·4116436	·0005230126
1913	3659569	7000755497	43·7378554	12·4138070	·0005227392
1914	3663396	7011739944	43·7492857	12·4159697	·0005224660
1915	3667225	7022735875	43·7607129	12·4181316	·0005221932
1916	3671056	7033743296	43·7721373	12·4202928	·0005219207
1917	3674889	7044762213	43·7835585	12·4224533	·0005216484
1918	3678724	7055792632	43·7949768	12·4246129	·0005213764
1919	3682561	7066834559	43·8063922	12·4267719	·0005211047
1920	3686400	7077888000	43·8178046	12·4289300	·0005208333
1921	3690241	7088952961	43·8292140	12·4310875	·0005205622
1922	3694084	7100029448	43·8406204	12·4332441	·0005202914
1923	3697929	7111117467	43·8520239	12·4354001	·0005200208
1924	3701776	7122217024	43·8634244	12·4375552	·0005197505
1925	3705625	7133328125	43·8748219	12·4397097	·0005194805
1926	3709476	7144450776	43·8862165	12·4418634	·0005192108
1927	3713329	7155584983	43·8976081	12·4440163	·0005189414
1928	3717184	7166730752	43·9089968	12·4461685	·0005186722
1929	3721041	7177888089	43·9203725	12·4483200	·0005184033
1930	3724900	7189057000	43·9317652	12·4504707	·0005181347
1931	3728761	7200237491	43·9431451	12·4526206	·0005178664
1932	3732624	7211429568	43·9545220	12·4547699	·0005175983
1933	3736489	7222633237	43·9658959	12·4569184	·0005173306
1934	3740356	7233848504	43·9772668	12·4590661	·0005170631
1935	3744225	7245075375	43·9886349	12·4612131	·0005167959
1936	3748096	7256313856	44·0000000	12·4633594	·0005165289
1937	3751969	7267563953	44·0113622	12·4655049	·0005162623
1938	3755844	7278825672	44·0227214	12·4676497	·0005159959
1939	3759721	7290099019	44·0340777	12·4697937	·0005157298
1940	3763600	7301384000	44·0454311	12·4719370	·0005154639
1941	3767481	7312680621	44·0567815	12·4740796	·0005151984
1942	3771364	7323988888	44·0681291	12·4762214	·0005149331
1943	3775249	7335308807	44·0794737	12·4783625	·0005146680
1944	3779136	7346640384	44·0908154	12·4805029	·0005144033
1945	3783025	7357983625	44·1021541	12·4826426	·0005141388
1946	3786916	7369338536	44·1134900	12·4847815	·0005138746
1947	3790809	7380705123	44·1248229	12·4869197	·0005136107
1948	3794704	7392083392	44·1361530	12·4890571	·0005133470
1949	3798601	7403473349	44·1474801	12·4911938	·0005130836
1950	3802500	7414875000	44·1588043	12·4933298	·0005128205
1951	3806401	7426288351	44·1701256	12·4954651	·0005125577
1952	3810304	7437713408	44·1814441	12·4975995	·0005122951
1953	3814209	7449150177	44·1927596	12·4997333	·0005120328
1954	3818116	7460598664	44·2040722	12·5018664	·0005117707
1955	3822025	7472058875	44·2153819	12·5039988	·0005115090
1956	3825936	7483530816	44·2266888	12·5061304	·0005112474

No.	Square	Cube	Square Root	Cube Root	Reciprocal
1957	3829849	7495014493	44·2379927	12·5082612	·0005109862
1958	3833764	7506509912	44·2492938	12·5103914	·0005107252
1959	3837681	7518017079	44·2605919	12·5125208	·0005104645
1960	3841600	7529536000	44·2718872	12·5146495	·0005102041
1961	3845521	7541066681	44·2831797	12·5167775	·0005099439
1962	3849444	7552609128	44·2944692	12·5189047	·0005096840
1963	3853369	7564163347	44·3057558	12·5210313	·0005094244
1964	3857296	7575729344	44·3170396	12·5231571	·0005091650
1965	3861225	7587307125	44·3283205	12·5252822	·0005089059
1966	3865156	7598896696	44·3395985	12·5274065	·0005086470
1967	3869089	7610498063	44·3508737	12·5295302	·0005083884
1968	3873024	7622111232	44·3621460	12·5316531	·0005081301
1969	3876961	7633736209	44·3734155	12·5337753	·0005078720
1970	3880900	7645373000	44·3846820	12·5358968	·0005076142
1971	3884841	7657021611	44·3959457	12·5380176	·0005073567
1972	3888784	7668682048	44·4072066	12·5401377	·0005070994
1973	3892729	7680354317	44·4184646	12·5422570	·0005068424
1974	3896676	7692038424	44·4297198	12·5443757	·0005065856
1975	3900625	7703734375	44·4409720	12·5464936	·0005063291
1976	3904576	7715442176	44·4522215	12·5486107	·0005060729
1977	3908529	7727161833	44·4634681	12·5507272	·0005058169
1978	3912484	7738893352	44·4747119	12·5528430	·0005055612
1979	3916441	7750636739	44·4859528	12·5549580	·0005053057
1980	3920400	7762392000	44·4971909	12·5570723	·0005050505
1981	3924361	7774159141	44·5084262	12·5591860	·0005047956
1982	3928324	7785938168	44·5196586	12·5612989	·0005045409
1983	3932289	7797729087	44·5308881	12·5634111	·0005042864
1984	3936256	7809531904	44·5421149	12·5655226	·0005040323
1985	3940225	7821346625	44·5533388	12·5676334	·0005037783
1986	3944196	7833173256	44·5645599	12·5697435	·0005035247
1987	3948169	7845011803	44·5757781	12·5718529	·0005032713
1988	3952144	7856862272	44·5869936	12·5739615	·0005030181
1989	3956121	7868724669	44·5982062	12·5760695	·0005027652
1990	3960100	7880599000	44·6094160	12·5781767	·0005025126
1991	3964081	7892485271	44·6206230	12·5802832	·0005022602
1992	3968064	7904383488	44·6318272	12·5823891	·0005020080
1993	3972049	7916293657	44·6430286	12·5844942	·0005017561
1994	3976036	7928215784	44·6542271	12·5865987	·0005015045
1995	3980025	7940149875	44·6654228	12·5887024	·0005012531
1996	3984016	7952095936	44·6766158	12·5908054	·0005010020
1997	3988009	7964053973	44·6878059	12·5929078	·0005007511
1998	3992004	7976023992	44·6989933	12·5950094	·0005005005
1999	3996001	7988005999	44·7101778	12·5971103	·0005002501
2000	4000000	8000000000	44·7213596	12·5992105	·0005000000
2001	4004001	8012006001	44·7325385	12·6013101	·0004997501
2002	4008004	8024024008	44·7437146	12·6034089	·0004995005
2003	4012009	8036054027	44·7548880	12·6055070	·0004992511
2004	4016016	8048096064	44·7660586	12·6076044	·0004990020
2005	4020025	8060150125	44·7772264	12·6097011	·0004987531

No.	Square	Cube	Square Root	Cube Root	Reciprocal
2006	4024036	8072216216	44·7883913	12·6117971	·0004985045
2007	4028049	8084294343	44·7995535	12·6138924	·0004982561
2008	4032064	8096384512	44·8107130	12·6159870	·0004980080
2009	4036081	8108486729	44·8218697	12·6180810	·0004977601
2010	4040100	8120601000	44·8330235	12·6201743	·0004975124
2011	4044121	813:727331	44·8441746	12·6222669	·0004972650
2012	4048144	8144865728	44·8553230	12·6243587	·0004970179
2013	4052169	8157016197	44·8664685	12·6264499	·0004967710
2014	4056196	8169178744	44·8776113	12·6285404	·0004965243
2015	4060225	8181353375	44·8887514	12·6306301	·0004962779
2016	4064256	8193540096	44·8998886	12·6327192	·0004060317
2017	4068289	8205738913	44·9110231	12·6348076	·0004957858
2018	4072324	8217949832	44·9221549	12·6368953	·0004955401
2019	4076361	8230172859	44·9332839	12·6389823	·0004952947
2020	4080400	8242408000	44·9444101	12·6410687	·0004950495
2021	4084441	8254655261	44·9555336	12·6431543	·0004948046
2022	4088484	8266914648	44·9666543	12·6452393	·0004945598
2023	4092529	8279186167	44·9777723	12·6473235	·0004943154
2024	4096576	8291469824	44·9888875	12·6494071	·0004940711
2025	4100625	8303765625	45·0000000	12·6514900	·0004938272
2026	4104676	8316073576	45·0111097	12·6535722	·0004935834
2027	4108729	8328393683	45·0222167	12·6556538	·0004933399
2028	4112784	8340725952	45·0333210	12·6577346	·0004930966
2029	4116841	8353070389	45·0444225	12·6598148	·0004928536
2030	4120900	8365427000	45·0555213	12·6618943	·0004926108
2031	4124961	8377795791	45·0666173	12·6639731	·0004923683
2032	4129024	8390176768	45·0777107	12·6660512	·0004921260
2033	4133089	8402569937	45·0888013	12·6681286	·0004918839
2034	4137156	8414975304	45·0998891	12·6702053	·0004916421
2035	4141225	8427392875	45·1109743	12·6722814	·0004914005
2036	4145296	8439822656	45·1220567	12·6743567	·0004911591
2037	4149369	8452264653	45·1331364	12·6764314	·0004909180
2038	4153444	8464718872	45·1442134	12·6785054	·0004906771
2039	4157521	8477185319	45·1552876	12·6805788	·0004904365
2040	4161600	8489664000	45·1663592	12·6826514	·0004901961
2041	4165681	8502154921	45·1774280	12·6847234	·0004899559
2042	4169764	8514658088	45·1884941	12·6867947	·0004897160
2043	4173849	8527173507	45·1995575	12·6888654	·0004894762
2044	4177936	8539701184	45·2106182	12·6909354	·0004892368
2045	4182025	8552241125	45·2216762	12·6930047	·0004889976
2046	4186116	8564793336	45·2327315	12·6950733	·0004887586
2047	4190209	8577357823	45·2437841	12·6971412	·0004885198
2048	4194304	8589934592	45·2548340	12·6992084	·0004882813
2049	4198401	8602523649	45·2658812	12·7012750	·0004880429
2050	4202500	8615125000	45·2769257	12·7033409	·0004878049
2051	4206601	8627738651	45·2879675	12·7054061	·0004875670
2052	4210704	8640364608	45·2990066	12·7074707	·0004873294
2053	4214809	8653002877	45·3100430	12·7095346	·0004870921
2054	4218916	8665653464	45·3210768	12·7115978	·0004868549

No.	Square	Cube	Square Root	Cube Root	Reciprocal
2055	4223025	8678316375	45·3321078	12·7136603	·0004866180
2056	4227136	8690991616	45·3431362	12·7157222	·0004863813
2057	4231249	8703679193	45·3541619	12·7177835	·0004861449
2058	4235364	8716379112	45·3651849	12·7198441	·0004859086
2059	4239481	8729091379	45·3762052	12·7219040	·0004856727
2060	4243600	8741816000	45·3872229	12·7239632	·0004854369
2061	4247721	8754552981	45·3982378	12·7260218	·0004852014
2062	4251844	8767302328	45·4092501	12·7280797	·0004849661
2063	4255969	8780064047	45·4202598	12·7301370	·0004847310
2064	4260096	8792838144	45·4312668	12·7321935	·0004844961
2065	4264225	8805624625	45·4422711	12·7342494	·0004842615
2066	4268356	8818423496	45·4532727	12·7363046	·0004840271
2067	4272489	8831234763	45·4642717	12·7383592	·0004837929
2068	4276624	8844058432	45·4752680	12·7404131	·0004835590
2069	4280761	8856894509	45·4862616	12·7424664	·0004833253
2070	4284900	8869743000	45·4972526	12·7445189	·0004830918
2071	4289041	8882603911	45·5082410	12·7465709	·0004828585
2072	4293184	8895477248	45·5192267	12·7486222	·0004826255
2073	4297329	8908363017	45·5302097	12·7506728	·0004823927
2074	4301476	8921261224	45·5411901	12·7527227	·0004821601
2075	4305625	8934171875	45·5521679	12·7547721	·0004819277
2076	4309776	8947094976	45·5631430	12·7568207	·0004816956
2077	4313929	8960030533	45·5741155	12·7588687	·0004814636
2078	4318084	8972978552	45·5850853	12·7609160	·0004812320
2079	4322241	8985939039	45·5960525	12·7629627	·0004810005
2080	4326400	8998912000	45·6070170	12·7650087	·0004807692
2081	4330561	9011897441	45·6179789	12·7670540	·0004805382
2082	4334724	9024895368	45·6289382	12·7690987	·0004803074
2083	4338889	9037905787	45·6398948	12·7711427	·0004800768
2084	4343056	9050928704	45·6508488	12·7731861	·0004798464
2085	4347225	9063964125	45·6618002	12·7752288	·0004796163
2086	4351396	9077012056	45·6727490	12·7772709	·0004793864
2087	4355569	9090072503	45·6836951	12·7793123	·0004791567
2088	4359744	9103145472	45·6946386	12·7813531	·0004789272
2089	4363921	9116230969	45·7055795	12·7833932	·0004786979
2090	4368100	9129329000	45·7165178	12·7854326	·0004784689
2091	4372281	9142439571	45·7274534	12·7874714	·0004782401
2092	4376464	9155562688	45·7383865	12·7895096	·0004780115
2093	4380649	9168698357	45·7493169	12·7915471	·0004777831
2094	4384836	9181846584	45·7602447	12·7935840	·0004775549
2095	4389025	9195007375	45·7711699	12·7956202	·0004773270
2096	4393216	9208180736	45·7820926	12·7976558	·0004770992
2097	4397409	9221366673	45·7930126	12·7996907	·0004768717
2098	4401604	9234565192	45·8039299	12·8017250	·0004766444
2099	4405801	9247776299	45·8148447	12·8037586	·0004764173
2100	4410000	9261000000	45·8257569	12·8057916	·0004761905
2101	4414201	9274236301	45·8366665	12·8078239	·0004759638
2102	4418404	9287485208	45·8475735	12·8098556	·0004757374
2103	4422609	9300746727	45·8584779	12·8118866	·0004755112

No.	Square	Cube	Square Root	Cube Root	Reciprocal
2104	4426816	9314020864	45·8693798	12·8139170	·0004752852
2105	4431025	9327307625	45·8802790	12·8159468	·0004750594
2106	4435236	9340607016	45·8911756	12·8179759	·0004748338
2107	4439449	9353919043	45·9020696	12·8200044	·0004746084
2108	4443664	9367243712	45·9129611	12·8220323	·0004743833
2109	4447881	9380581029	45·9238500	12·8240595	·0004741584
2110	4452100	9393931000	45·9347363	12·8260861	·0004739336
2111	4456321	9407293631	45·9456200	12·8281120	·0004737091
2112	4460544	9420668928	45·9565012	12·8301378	·0004734848
2113	4464769	9434056897	45·9673798	12·8321620	·0004732608
2114	4468996	9447457544	45·9782557	12·8341860	·0004730369
2115	4473225	9460870875	45·9891291	12·8362094	·0004728132
2116	4477456	9474296896	46·0000000	12·8382321	·0004725898
2117	4481689	9487735613	46·0108683	12·8402542	·0004723666
2118	4485924	9501187032	46·0217340	12·8422756	·0004721435
2119	4490161	9514651159	46·0325971	12·8442964	·0004719207
2120	4494400	9528128000	46·0434577	12·8463166	·0004716981
2121	4498641	9541617561	46·0543158	12·8483361	·0004714757
2122	4502884	9555119848	46·0651712	12·8503551	·0004712535
2123	4507129	9568634867	46·0760241	12·8523733	·0004710316
2124	4511376	9582162624	46·0868745	12·8543910	·0004708098
2125	4515625	9595703125	46·0977223	12·8564080	·0004705882
2126	4519876	9609256376	46·1085675	12·8584243	·0004703669
2127	4524129	9622822383	46·1194102	12·8604401	·0004701457
2128	4528384	9636401152	46·1302504	12·8624552	·0004699248
2129	4532641	9649992689	46·1410880	12·8644697	·0004697041
2130	4536900	9663597000	46·1519230	12·8664835	·0004694836
2131	4541161	9677214091	46·1627555	12·8684967	·0004692633
2132	4545424	9690843968	46·1735855	12·8705093	·0004690432
2133	4549689	9704486637	46·1844130	12·8725213	·0004688233
2134	4553956	9718142104	46·1952378	12·8745326	·0004686036
2135	4558225	9731810375	46·2060602	12·8765433	·0004683841
2136	4562496	9745491456	46·2168800	12·8785534	·0004681648
2137	4566769	9759185353	46·2276973	12·8805628	·0004679457
2138	4571044	9772892072	46·2385121	12·8825717	·0004677268
2139	4575321	9786611619	46·2493243	12·8845199	·0004675082
2140	4579600	9800344000	46·2601340	12·8865874	·0004672897
2141	4583881	9814089221	46·2709412	12·8885944	·0004670715
2142	4588164	9827847288	46·2817459	12·8906007	·0004668534
2143	4592449	9841618207	46·2925480	12·8926064	·0004666356
2144	4596736	9855401984	46·3033476	12·8946115	·0004664179
2145	4601025	9869198625	46·3141447	12·8966159	·0004662005
2146	4605316	9883008136	46·3249393	12·8986197	·0004659832
2147	4609609	9896830523	46·3357314	12·9006229	·0004657662
2148	4613904	9910665792	46·3465209	12·9026255	·0004655493
2149	4618201	9924513949	46·3573079	12·9046275	·0004653327
2150	4622500	9938375000	46·3680924	12·9066288	·0004651163
2151	4626801	9952248951	46·3788745	12·9086295	·0004649000
2152	4631104	9966135808	46·3896540	12·9106296	·0004646840

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No.	Square	Cube	Square Root	Cube Root	Reciprocal
2153	4635409	9980035577	46·4004310	12·9126291	·0004644682
2154	4639716	9993948264	46·4112055	12·9146279	·0004642526
2155	4644025	10007873875	46·4219775	12·9166262	·0004640371
2156	4648336	10021812416	46·4327471	12·9186238	·0004638219
2157	4652649	10035763893	46·4435141	12·9206208	·0004636069
2158	4656964	10049728312	46·4542786	12·9226172	·0004633920
2159	4661281	10063705679	46·4650406	12·9246129	·0004631774
2160	4665600	10077696000	46·4758002	12·9266081	·0004629630
2161	4669921	10091699281	46·4865572	12·9286027	·0004627487
2162	4674244	10105715528	46·4973118	12·9305966	·0004625347
2163	4678569	10119744747	46·5080638	12·9325899	·0004623209
2164	4682896	10133786944	46·5188134	12·9345827	·0004621072
2165	4687225	10147842125	46·5295605	12·9365747	·0004618938
2166	4691556	10161910296	46·5403051	12·9385662	·0004616805
2167	4695889	10175991463	46·5510472	12·9405570	·0004614675
2168	4700224	10190085632	46·5617869	12·9425472	·0004612546
2169	4704561	10204192809	46·5725241	12·9445369	·0004610420
2170	4708900	10218313000	46·5832588	12·9465259	·0004608295
2171	4713241	10232446211	46·5939910	12·9485143	·0004606172
2172	4717584	10246592448	46·6047208	12·9505021	·0004604052
2173	4721929	10260751717	46·6154481	12·9524893	·0004601933
2174	4726276	10274924024	46·6261729	12·9544759	·0004599816
2175	4730625	10289109375	46·6368953	12·9564618	·0004597701
2176	4734976	10303307776	46·6476152	12·9584472	·0004595588
2177	4739329	10317519233	46·6583326	12·9604319	·0004593477
2178	4743684	10331743752	46·6690476	12·9624161	·0004591368
2179	4748041	10345981339	46·6797601	12·9643996	·0004589261
2180	4752400	10360232000	46·6904701	12·9663826	·0004587156
2181	4756761	10374495741	46·7011777	12·9683649	·0004585053
2182	4761124	10388772568	46·7118829	12·9703466	·0004582951
2183	4765489	10403062487	46·7225855	12·9723277	·0004580852
2184	4769856	10417365504	46·7332858	12·9743082	·0004578755
2185	4774225	10431681625	46·7439836	12·9762881	·0004576659
2186	4778596	10446010856	46·7546789	12·9782674	·0004574565
2187	4782969	10460353203	46·7653718	12·9802461	·0004572474
2188	4787344	10474708672	46·7760623	12·9822242	·0004570384
2189	4791721	10489077269	46·7867503	12·9842017	·0004568296
2190	4796100	10503459000	46·7974358	12·9861786	·0004566210
2191	4800481	10517853871	46·8081189	12·9881549	·0004564126
2192	4804864	10532261888	46·8187996	12·9901306	·0004562044
2193	4809249	10546683057	46·8294779	12·9921057	·0004559964
2194	4813636	10561117384	46·8401537	12·9940802	·0004557885
2195	4818025	10575564875	46·8508271	12·9960540	·0004555809
2196	4822416	10590025536	46·8614981	12·9980273	·0004553734
2197	4826809	10604499373	46·8721666	13·0000000	·0004551661
2198	4831204	10618986392	46·8828327	13·0019721	·0004549591
2199	4835601	10633486599	46·8934963	13·0039436	·0004547522
2200	4840000	10648000000	46·9041576	13·0059145	·0004545455
2201	4844401	10662526601	46·9148164	13·0078848	·0004543389

Table CXCVII.—Fourth Powers of Numbers.

No.	4th Power.	No.	4th Power.	No.	4th Power.	No.	4th Power.
1	1	26	456,976	51	6,765,201	76	33,362,176
2	16	27	531,441	52	7,311,616	77	35,153,041
3	81	28	614,656	53	7,890,481	78	37,015,056
4	256	29	707,281	54	8,503,056	79	38,950,081
5	625	30	810,000	55	9,150,625	80	40,960,000
6	1,296	31	923,521	56	9,834,496	81	43,046,721
7	2,401	32	1,048,576	57	10,556,001	82	45,212,176
8	4,096	33	1,185,921	58	11,316,496	83	47,458,321
9	6,561	34	1,336,336	59	12,117,361	84	49,787,136
10	10,000	35	1,500,625	60	12,960,000	85	52,200,625
11	14,641	36	1,679,616	61	13,845,841	86	54,708,016
12	20,736	37	1,874,161	62	14,776,336	87	57,289,761
13	28,561	38	2,085,136	63	15,752,961	88	59,969,536
14	38,416	39	2,313,441	64	16,777,216	89	62,742,241
15	50,625	40	2,560,000	65	17,850,625	90	65,610,000
16	65,536	41	2,825,761	66	18,974,736	91	68,574,961
17	83,521	42	3,111,696	67	20,151,121	92	71,639,296
18	104,976	43	3,418,801	68	21,381,376	93	74,805,201
19	130,321	44	3,748,096	69	22,667,121	94	78,074,896
20	160,000	45	4,100,625	70	24,010,000	95	81,450,625
21	194,481	46	4,477,456	71	25,411,681	96	84,934,656
22	234,256	47	4,879,681	72	26,873,856	97	88,529,281
23	279,841	48	5,308,416	73	28,398,241	98	92,236,816
24	331,776	49	5,764,801	74	29,986,576	99	96,059,601
25	390,625	50	6,250,000	75	31,640,625	100	100,000,000

Table CXCVIIa.—Fourth Root of Numbers.

No.	4th Root.	No.	4th Root.	No.	4th Root.	No.	4th Root.
1	1·000	13	1·899	25	2·236	37	2·466
2	1·189	14	1·934	26	2·258	38	2·482
3	1·316	15	1·968	27	2·279	39	2·499
4	1·414	16	2·000	28	2·300	40	2·515
5	1·495	17	2·031	29	2·320	41	2·530
6	1·565	18	2·060	30	2·340	42	2·546
7	1·626	19	2·088	31	2·360	43	2·560
8	1·682	20	2·115	32	2·378	44	2·574
9	1·732	21	2·141	33	2·397	45	2·590
10	1·778	22	2·166	34	2·415	46	2·605
11	1·821	23	2·190	35	2·432	47	2·618
12	1·861	24	2·215	36	2·449	48	2·631

Table CXCVIIb.—Fourth Powers of Shaft Diameters.

Dia.	4th Power.	Dia.	4th Power.	Dia.	4th Power.	Dia.	4th Power.	Dia.	4th Power.
1	1·00	4⅜	366	7⅜	3,607	12	20,736	18½	117,160
1⅛	1·60	4½	410	7⅞	3,846	12¼	22,500	18¾	123,600
1¼	2·44	4⅝	458	8	4,096	12½	24,430	19	130,321
1⅜	3·57	4¾	509	8⅛	4,358	12¾	26,439	19¼	137,400
1½	5·06	4⅞	565	8¼	4,632	13	28,561	19½	144,600
1⅝	6 97	5	625	8⅜	4,920	13¼	30,835	19¾	152,100
1¾	9·38	5⅛	690	8½	5,220	13½	33,233	20	160,000
1⅞	12·4	5¼	760	8⅝	5,534	13¾	35,759	20¼	168,100
2	16·0	5⅜	835	8¾	5,862	14	38,416	20½	176,400
2⅛	20·4	5½	915	8⅞	6,204	14¼	41,250	20¾	185,400
2¼	25·6	5⅝	1,001	9	6,561	14½	44,226	21	194,480
2⅜	31·8	5¾	1,093	9⅛	6,922	14¾	47,350	21¼	203,800
2½	39·1	5⅞	1,191	9¼	7,321	15	50,625	21½	213,540
2⅝	47·5	6	1,296	9⅜	7,725	15¼	54,100	21¾	223,730
2¾	57·2	6⅛	1,407	9½	8,145	15½	57,750	22	234,256
2⅞	68·3	6¼	1,526	9⅝	8,582	15¾	61,520	22¼	245,025
3	81·0	6⅜	1,652	9¾	9,037	16	65,536	22½	256,036
3⅛	95·4	6½	1,785	9⅞	9,509	16¼	69,690	22¾	268,320
3¼	112	6⅝	1,926	10	10,000	16½	74,110	23	279,840
3⅜	130	6¾	2,076	10¼	11,025	16¾	78,720	23¼	292,680
3½	150	6⅞	2,234	10½	12,166	17	83,521	23½	304,700
3⅝	173	7	2,401	10¾	13,363	17¼	88,560	23¾	318,100
3¾	198	7⅛	2,577	11	14,641	17½	93,850	24	331,380
3⅞	225	7¼	2,763	11¼	16,027	17¾	99,225	24¼	345,740
4	256	7⅜	2,958	11½	17,503	18	104,970	24½	360,000
4⅛	290	7½	3,164	11¾	19,072	18¼	110,930	24¾	375,770
4¼	326	7⅝	3,380						

Table CXCVIII.—Tons of Water delivered per hour with a Loss of Head of 5 lbs. through Pipes of various Sizes and Lengths $\left(Q=10\sqrt{\frac{d^5\times 5}{L}}\right)$.

Bore of Pipe.	Length of Pipe in feet.													
	5	10	15	20	30	40	50	60	70	80	100	120	140	160
ins.														
1	5·00	3·54	2·89	2·50	2·04	1·77	1·57	..	..	..	..	..	..	..
1½	23·1	19·5	15·9	13·8	11·3	9·75	8·76	..	..	..	..	..	..	..
2	56·5	40·0	32·8	28·4	21·4	20·1	18·1	..	..	..	..	..	..	..
2½	98·8	70·0	57·1	50·0	40·3	35·0	31·2	28·5	26·4	..	..	..	..	..
3	156	110	90·0	77·9	63·6	55·2	49·3	45·0	41·6	..	..	..	..	..
3½	229	162	133	115	93·5	81·0	72·4	66·1	61·2	57·3	51·2	..	..	..
4	320	227	185	160	131	113	101	92·4	85·5	80·0	71·6	65·3	60·5	56·6
4½	430	304	248	215	175	152	136	124	115	108	96·1	87·8	81·2	76·0
5	562	400	325	281	229	200	177	162	150	140	126	115	106	100
6	882	624	509	441	360	312	279	255	236	221	197	180	167	156
7	1,298	909	749	649	529	459	411	374	346	324	290	265	245	230
8	1,811	1,281	1,047	906	739	641	573	523	484	453	405	370	342	320
9	2,433	1,702	1,406	1194	993	861	770	702	650	609	544	497	460	430
10	3,119	2,237	1,827	1582	1290	1119	1000	912	845	791	707	646	600	559
12	4,989	3,530	2,883	2495	2036	1765	1578	1439	1332	1248	1116	1019	943	882
14	7,335	5,190	4,239	3670	3000	2600	2320	2116	1960	1834	1640	1500	1386	1300
16	10,241	7,247	5,919	5124	4180	3623	3253	2955	2733	2561	2290	..	..	..
18	13,305	9,730	7,946	6880	5611	4865	4350	4000	3674	..	..	..	..	..
20	17,900	12,660	10,336	9000	7300	6330	5658	..	..	..	..	..	..	..

N.B.—The amount delivered by any other head p_1 can be found by multiplying the above by $\sqrt{0\cdot2\times p_1}$ or $0\cdot45\sqrt{p_1}$.

Table CXCIX.—Hyperbolic Logarithms of Numbers up to 232.

No.	Logthm.	No.	Logthm.	No.	Logthm.	No.	Logthm.
1·10	0·0953	1·60	0·4700	2·20	0·7885	3·20	1·1632
1·15	0·1398	1·65	0·5008	2·30	0·8329	3·30	1·1939
1·20	0·1823	1·70	0·5306	2·40	0·8755	3·40	1·2238
1·25	0·2231	1·75	0·5596	2·50	0·9163	3·50	1·2528
1·30	0·2624	1·80	0·5878	2·60	0·9555	3·60	1 2809
1·35	0·3001	1·85	0·6152	2·70	0·9932	3·70	1·3083
1·40	0·3365	1·90	0·6419	2·80	1·0296	3·80	1·3350
1·45	0·3720	1·95	0·6678	2·90	1·0647	3·90	1·3610
1·50	0·4055	2·0	0·6931	3·00	1·0986	4·00	1·3863
1·55	0·4382	2·10	0·7419	3·10	1·1314	4·20	1·4351

Table CXCIX.—Hyperbolic Logarithms—*continued.*

No.	Logthm.	No.	Logthm.	No.	Logthm.	No.	Logthm.
4·40	1·4816	24	3·1781	66	4·1898	109	4·6921
4·60	1·5261	25	3·2189	67	4·2050	110	4·7013
4·80	1·5686	26	3·2581	68	4·2197	111	4·7103
5·00	1·6094	27	3·2958	69	4·2343	112	4·7193
5·20	1·6487	28	3·3322	70	4·2487	113	4·7283
5·40	1·6864	29	3·3673	71	4·2630	114	4·7370
5·60	1·7228	30	3·4012	72	4·2768	115	4·7458
5·80	1·7519	31	3·4341	73	4·2907	116	4·7555
6·00	1·7918	32	3·4656	74	4·3043	117	4·7630
6·20	1·8245	33	3·4965	75	4·3178	118	4·7715
6·40	1·8563	34	3·5267	76	4·3310	119	4·7798
6·60	1·8871	35	3·5557	78	4·3568	120	4·7883
6·80	1·9094	36	3·5838	79	4·3696	121	4·7966
7·00	1·9459	37	3·6111	80	4·3823	122	4·8049
7·20	1·9741	38	3·6379	81	4·3947	123	4·8130
7·40	2·0015	39	3·6639	82	4·4069	124	4·8211
7·60	2·0281	40	3·6893	83	4·4191	125	4·8291
7·80	2·0541	41	3·7139	84	4·4311	126	4·8372
8·00	2·0794	42	3·7378	85	4·4429	127	4·8450
8·20	2·1041	43	3·7615	86	4·4546	128	4·8528
8·40	2·1282	44	3·7849	87	4·4661	129	4·8607
8·60	2·1518	45	3·8069	88	4·4781	130	4·8683
8·80	2·1748	46	3·8290	89	4·4894	131	4·8761
9·00	2·1972	47	3·8504	90	4·5005	132	4·8837
9·20	2·2192	48	3·8714	91	4·5115	133	4·8913
9·40	2·2407	49	3·8921	92	4·5225	134	4·8987
9·60	2·2618	50	3·9123	93	4·5334	135	4·9060
9·80	2·2824	51	3·9321	94	4·5440	136	4·9134
10·0	2·3026	52	3·9515	95	4·5546	137	4·9208
11·0	2·3979	53	3·9706	96	4·5652	138	4·9281
12·0	2·4849	54	3·9893	97	4·5756	139	4·9353
13·0	2·5649	55	4·0077	98	4·5857	140	4·9424
14·0	2·6391	56	4·0256	99	4·5958	141	4·9496
15	2·7081	57	4·0433	100	4·6052	142	4·9567
16	2·7726	58	4·0606	101	4·6159	143	4·9638
17	2·8332	59	4·0779	102	4·6258	144	4·9708
18	2·8904	60	4·0947	103	4·6354	145	4·9777
19	2·9444	61	4·1111	104	4·6451	146	4·9846
20	2·9957	62	4·1274	105	4·6548	147	4·9914
21	3·0445	63	4·1433	106	4·6642	148	4·9982
22	3·0911	64	4·1592	107	4·6737	149	5·0049
23	3·1353	65	4·1746	108	4·6829	150	5·0115

Table CXCIX.—Hyperbolic Logarithms—*continued.*

No.	Logthm.	No.	Logthm.	No.	Logthm.	No.	Logthm.
151	5·0181	172	5·1483	193	5·2636	213	5·3622
152	5·0246	173	5·1541	194	5·2688	214	5·3669
153	5·0312	174	5·1598	195	5·2740	215	5·3716
154	5·0378	175	5·1656	196	5·2791	216	5·3763
155	5·0443	176	5·1714	197	5·2842	217	5·3809
156	5·0507	177	5·1771	198	5·2893	218	5·3855
157	5·0572	178	5·1827	199	5·2943	219	5·3900
158	5·0636	179	5·1883	200	5·2992	220	5·3945
159	5·0698	180	5·1939	201	5·3043	221	5·3991
160	5·0760	181	5·1994	202	5·3093	222	5·4037
161	5·0822	182	5·2050	203	5·3142	223	5 4081
162	5·0884	183	5·2104	204	5·3190	224	5·4125
163	5·0945	184	5 2158	205	5·3240	225	5·4171
164	5·1006	185	5·2212	206	5·3289	226	5·4216
165	5·1068	186	5 2266	207	5·3337	227	5·4259
166	5·1129	187	5·2320	208	5·3385	228	5·4302
167	5·1189	188	5·2374	209	5·3433	229	5·4346
168	5·1248	189	5·2427	210	5·3480	230	5·4390
169	5·1307	190	5·2480	211	5·3522	231	5·4434
170	5·1366	191	5·2532	212	5·3574	232	5·4477
171	5·1425	192	5·2584				

Table CXCIXa.—Power Transmitted per Revolution through Standard Shafts by Rule diameter $= \sqrt[3]{\frac{\text{S.H.P.}}{\text{R}}} \times 64.$

Diameter of Shaft	S.H.P. ÷ R.	Diameter of Shaft.	S.H.P. ÷ R.	Diameter of Shaft.	S.H.P. ÷ R.	Diameter of Shaft.	S.H.P. ÷ R.
ins.		ins		ins.		ins.	
5·00	1·953	8·0	8·000	12·0	27·00	18·0	91·25
5·25	2·262	8·25	8·766	12·5	30·52	18·5	98·92
5·50	2·600	8·50	9·594	13·0	34·32	19·0	107·2
5·75	2·970	8·75	10·47	13·5	38·44	19·5	115·8
6·00	3·375	9·00	11·41	14·0	43·25	20·0	125·0
6·25	3·814	9·25	12·36	14·5	47·62	20·5	134·6
6·50	4·291	9·50	13·39	15·0	52·74	21·0	144·7
6·75	4·805	9·75	14·49	15·5	58·19	21·5	155·2
7·0	5·363	10·0	15·62	16·0	64·00	22·0	166·4
7·25	5·954	10·5	18·08	16·5	70·19	22·5	178·0
7·5	6·591	11·0	20·80	17·0	76·76	23·0	190·1
7·75	7·274	11·5	23·77	17·5	83·74	24·0	216·0

NOMENCLATURE AND DEFINITIONS AS ESTABLISHED BY THE BRITISH MARINE ENGINEERING DESIGN AND CONSTRUCTION COMMITTEE.

(A) Boilers: Definitions of Various Sorts.

A Main Boiler is one whose special and general function is to supply steam to the engines engaged in driving the propellers of the ship, or to the auxiliaries necessary for the proper working of the same, and whose normal supply of feed water is from a surface condenser.

An Auxiliary Boiler is one used generally for purposes other than that of supplying the main propelling machinery, and, while it may be so used on an emergency, it is not a necessary part of the main boiler installation as used when at full service speed. It is, however, supplied with feed water from a surface condenser, or with other water equally pure, except on emergency.

Winch (Donkey) Boilers are those having no connection with the main boilers but employed on winches, cranes, and other appliances generally outside the engine- and boiler-rooms, and whose normal supply of feed water is from the sea or other impure water which will deposit solid matter in the boiler.

Vertical Boilers are those whose cylindrical shells are placed with the axis vertical or nearly so, and have the fire grate in the furnace at or near to the bottom.

(B) Tubes and Pipes: Definitions of Terms used.

A straight hollow cylindrical body made of copper, brass, iron, steel, or other malleable metal shall be called a *Tube.*

A tube when bent, flanged or otherwise worked and treated mechanically, or made up for service shall be called a *Pipe.*

The thickness of all tubes and pipes over 18 L.S.G. thick shall be expressed in *100ths of an inch.*

(C) Shafting: Nomenclature and Definitions of.

1. *Line of Shafting.*—A line of shafting in a screw steamship consists of the motor shaft at one end and the shaft to which the propeller is attached at the other, together with the shafts intermediate between them.

2. *Motor Shaft.* — The motor shaft in the case of reciprocating engines is the crank shaft, and in turbine engines is the rotor shaft. With an electrical drive it is the rotor or armature shaft.

3. *Tail Shaft.*—The shaft to which the propeller is fitted shall be called the tail shaft.

4. *Tube Shaft.*—When the tail shaft is entirely outboard, the shaft which passes into the inside of the ship shall be called the tube shaft.

5. *Outboard Intermediate Shaft.*—When there is a shaft interposed between the tail shaft and the tube shaft, it shall be called the outboard intermediate shaft.

6. *Intermediate Shafts.*—The shafts connecting the tube shaft with the motor shaft shall be called intermediate shafts.

7. *Thrust Shaft.*—The shaft having collars formed upon it for the purpose of resisting the longitudinal thrust on the shaft shall be called the thrust shaft.

8. *Couplings.*—The flanges forged solid with a shaft, or welded to the body as with iron and scrap steel shafts, and the other portions forged with, or those which are fitted to the shafts and are removable, whereby the shaft may be connected to other shafts, shall be called couplings.

9. *Coupling Bolts.*—When two shafts are connected together by fitted bolts with heads or with taper bodies, secured in place by nuts, they shall be called coupling bolts.

Driver Bolts.—Those bolts having plain cylindrical bodies without heads, which are secured to the coupling of one shaft by nuts, and are free to move longitudinally in the holes of the coupling of the adjacent shaft, shall be called driver bolts.

10. With geared turbines the shafts having on them the pinions and gear wheels, shall be called as follows:—

Motor Spindle.—Motor spindle or motor shaft is the axle or shaft of the rotor of a turbine or electric motor.

First Pinion Spindle.—The first pinion spindle is that coupled to the motor spindle and having on it the pinion which gears into the wheel of the next spindle or shaft.

Second Pinion Spindle.—When there is double gearing the second pinion spindle is that which has on it the wheel driven by the first pinion and also the second pinion which drives the wheel on the next spindle or shaft.

Main Wheel Shaft.—The main wheel shaft is that which carries the final wheel and is coupled to the intermediate shafting.

When there are more than one set of gearing provided to each line of shafting on account of there being more than one turbine to each, these spindles will have added to their designations high pressure, medium, or low pressure (H.P., M.P., L.P.).

11. *Crank Webs.*—In the case of cranks shafts, the arms shall be called crank webs.

Table CC.—Horse-power Transmissible by Solid Steel Shafts from Turbines, Electric, or other Motors having Uniform Torque (B.M.E.D. & C. Committee Rule, $d^3 = S.H.P. \times 64 \div Revs.\ p.m.$).

Diameter of Shaft.	Revolutions per Minute.													
	75.	100.	125.	150.	175.	200.	225.	250.	275.	300.	325.	350.	375.	400.
ins.														
5·0	146	195	244	293	342	391	439	488	537	586	635	683	732	781
5·5	195	260	325	390	455	520	585	650	715	780	845	910	975	1,040
6·0	254	339	424	509	594	679	764	849	933	1,018	1,103	1,188	1,273	1,358
6 5	322	429	536	644	751	858	966	1,073	1,180	1,287	1,395	1,502	1,610	1,717
7·0	402	536	670	804	938	1,072	1,206	1,340	1,474	1,608	1,742	1,876	2,010	2,144
7·5	495	660	825	990	1,155	1,320	1,485	1,650	1,815	1,980	2,145	2,310	2,475	2,640
8·0	600	800	1,000	1,200	1,400	1,600	1,800	2,000	2,200	2,400	2,600	2,800	3,000	3,200
8·5	720	960	1,200	1,440	1,680	1,920	2,160	2,400	2,640	2,880	3,120	3,360	3,600	3,840
9·0	854	1,140	1,425	1,709	1,994	2,279	2,564	2,848	3,134	3,418	3,702	3,988	4,272	4,556
9·5	1,000	1,334	1,667	2,001	2,334	2,668	3,002	3,335	3,668	4,002	4,335	4,669	5,002	5,336
10·0	1,170	1,560	1,950	2,340	2,730	3,120	3,510	3,900	4,290	4,680	5,070	5,460	5,850	6,240
10·5	1,356	1,808	2,260	2,712	3,164	3,616	4,068	4,520	4,972	5,424	5,876	6,328	6,780	7,232
11·0	1,560	2,080	2,600	3,120	3,640	4,160	4,680	5,200	5,720	6,240	6,760	7,280	7,800	8,320
11·5	1,785	2,380	2,975	3,570	4,165	4,760	5,355	5,950	6,545	7,130	7,725	8,320	8,915	9,510
12·0	2,025	2,700	3,375	4,050	4,725	5,400	6,075	6,750	7,425	8,100	8,775	9,450	10,125	10,800
13	2,574	3,432	4,290	5,148	6,006	6,864	7,722	8,580	9,438	10,296	11,154	12,012	12,870	13,728
14	3,216	4,288	5,360	6,432	7,504	8,576	9,648	10,720	11,792	12,864	13,936	15,008	16,080	17,152
15	3,954	5,272	6,590	7,908	9,226	10,544	11,862	13,180	14,498	15,816	17,134	18,452	19,770	21,088
16	4,800	6,400	8,000	9,600	11,200	12,800	14,400	16.000	17,600	19,200	20,800	22,400	24,000	25,600
17	5,757	7,676	9,595	11,514	13,433	15,352	17,271	19,190	21,109	23,028	24,947	26,866	28,785	30,700
18	6,834	9,112	11,390	13,668	15,946	18,224	20,502	22,780	25,058	27,336	29,614	31,892	34,170	36,448
19	8,040	10,720	13,400	16,080	18,760	21,440	24,120	26,800	29,480	32,160	34.840	37,520	40,200	42.880
20	9,375	12,500	15,625	18,750	21,875	25,000	28,125	31,250	34,375	37,500	40,625	43,750	46,875	50,000

N.B.—If from a two-crank 90° engine, 70 per cent. should be the limit. From a three-crank 120° or a balanced four-crank, 78 per cent.; and from a four-crank 90°, 75 per cent. For smooth water service, 110 per cent. each.

Table CCI.—Insulating Materials for Clothing Boilers, Steam Cylinders and Pipes (12 inches thick).

Air-dried Materials.	B.T.U. per sq. ft. per 1° F. per hour.	Weight of Medium per sq. ft.	Specific Heat.	Cost per Cub. foot.
		lbs.		pence.
1. Slag wool (light) . .	0·054	8·6	0·17	8·3
2. Hair felt . . .	0·058	7·9	...	20·6
3. Light magnesia . .	0·062	10·1	...	26·0
4. Granulated cork . .	0·069	6·1	0·43	9·8
5. Slag wool (heavy) . .	0·070	32·9	...	31·8
6. Kieselguhr . . .	0·073	15·0	0·23	9·6
7. Flaky charcoal . .	0·082	14·5	0·29	7·8
8. Pumice ($\frac{1}{8}$-in. mean dia.)	0·095	25·0	...	24·1
9. Sawdust spruce . .	0·096	13·1	...	0·5
10. Asbestos fibre . . .	0·136	14·5	...	137·0

Table CCII.—Refractory Materials for Boilers.

(Temperature in water-tube boiler furnaces, 3000° F.)

Material.	Degrees.	
	Centigrade.	Fahrenheit.
Best fire bricks, melting point . .	1799	3270
Inferior fire bricks, ,, . .	1530	2820
,, ,, soften at . . .	1399	2550
Silica bricks, ,, . . .	1530	2820
,, melt at	1670	3090
Bauxite clay, melts at	1793	3260
Pure silica, ,, ...	1730	3180
Silicon, ,, ...	1421	2590

APPENDIX A.

RULES OF THE BRITISH CORPORATION FOR THE SURVEY AND REGISTRY OF SHIPPING (MACHINERY).

GENERAL CONDITIONS OF CLASSIFICATION.

(Section 31.)

The construction of the machinery for Steam Vessels, and of donkey boilers for Sailing Vessels, which are intended for classification with the British Corporation, is to be carried out under the supervision and to the satisfaction of the Surveyors, and before the work is proceeded with, detailed plans and particulars of the boilers and superheaters, copies of the specifications of boiler material, and full information regarding the engines, together with plans showing proposed arrangement of engine seating and holding-down bolts, plans of bilge and ballast pumping arrangements, evaporators, feed-water filters and heaters, and particulars of sizes of steam and feed pipes, are to be submitted for the approval of the Committee. The clear space between bottom of boilers and top of tank of floors to be stated in all cases, and is not to be less than 18 inches, without the special sanction of the Committee. Upon satisfactory completion in accordance with the Rules, the machinery will be entered in the Register Book "M.B.S.*" (Machinery British Standard—Special Survey).

BOILERS. (Section 32.)

This Registration Society now accepts and uses the new Rules of the Board of Trade.

1. The Rules are intended to apply to the construction of steel boilers; where boilers are to be made of iron, they will be specially considered by the Committee.

2. The quality of steel to be used in the construction of boilers must be of the best "mild" quality, made on the "Open Hearth" system, free from hammer-dressing, cracks, and surface defects. It must have a workmanlike finish, and be capable of standing the following tests:—

3. All samples shall be selected by the Surveyor, properly marked, and tested in his presence prior to despatch, and the tensile strength and ductility shall be determined from test pieces cut lengthwise or crosswise from the rolled material, in accordance with the requirements specified in Sec. 3, par. 3:—

The tensile strength and ductility shall be determined from test pieces cut lengthwise or crosswise from the finished material, and when material is annealed or otherwise treated before despatch the test pieces shall be similarly and simultaneously treated before testing. Wherever practicable the rolled surfaces shall be retained on two opposite sides of the test piece, and any straightening of test pieces which may be required shall be done cold. The parallel part of the tensile test piece shall not be less than 9 inches in length, and the width shall not exceed 1½ inches for material more than $\frac{3}{4}$ inch in thickness, from $\frac{3}{4}$ to $\frac{1}{2}$ it shall not exceed 2 inches, and under $\frac{1}{2}$ the width shall not be more than 2½ inches. For round bars the parallel part may be either nine or four times the diameter of the test piece, but the sectional area must not be less than ¼ square inch; the bars may be tested the full size as rolled. Should a tensile test piece break outside the middle half of its gauge length the test may, at the maker's option, be discarded and another test made of the same plate or bar. Test pieces for bends shall not be less than 1½ inches wide, and for small bars the whole section may be used. For temper bend tests the samples are to be heated uniformly to a blood red and quenched in water at a temperature not exceeding 80° Fahrenheit. The colour shall be judged indoors in the shade. The rough edge or arris caused by shearing may be removed from all cold bends and from temper bends above $\frac{1}{2}$ inch in thickness by filing or grinding, pieces 1 inch thick and above may be machined, but the test pieces shall receive no other preparation. The pieces are not to be annealed unless the material from which they are cut is similarly and simultaneously treated.

4. One tensile test and one temper or cold bend test shall be taken from each plate as rolled. For plates exceeding 2½ tons in weight one tensile and one bend test shall be taken from each end—one bend test to be temper and the other cold. For angle bars, rivet bars, and stay bars at least two tensile tests shall be taken from each charge; but when the number of the bars, as rolled, from one charge exceeds 15, an additional tensile test shall be made for each batch of 15 bars, or portion thereof, with the exception of round bars 1¾ inch diameter and under, when the number shall be 50 in place of 15. A temper or cold bend test shall be made from each angle bar as rolled, a temper and cold bend test shall be made from every fifteen stay bars as rolled, from each charge, but for rivet bars no bend tests will be required.

5 The tensile breaking strength of plates for shells shall be between the limits of 28 and 32 tons per square inch if not otherwise specified. For plates intended for flanging or welding and for combustion chambers and furnaces, also for bars for combustion chamber stays

and rivet bars, the tensile breaking strength shall be between the limits of 26 and 30 tons per square inch. In the case of material for purposes in which tensile strength is not important, the tensile test may be dispensed with, and the bend test only made if so specified and approved.

All material which has satisfactorily passed the requirements must be clearly stamped by the manufacturer in at least two places on each finished bar or plate, thus—B.C., signifying that the material has satisfactorily complied with the tests above described. Material the tensile breaking strength of which is between the limits of 26 and 30 tons, and complies with the corresponding requirements for elongation, should be marked $\frac{\text{B.C.}}{\text{F.}}$ to signify that it is of "cold flanging" quality, and material which has been approved on bend tests only is to be marked $\frac{\text{B.C.}}{\text{B.}}$ so that it may be readily identified.

No material bearing these brands is to be forwarded from the steel works until the prescribed tests have been made by the Surveyors and the mill sheets signed by them. The material must also be legibly stamped with the manufacturer's name or trade mark, and the place where made.

6. The elongation on a gauge length of 8 inches or 8 diameters shall not be less than 20 per cent. for material required to have a tensile breaking strength of 28 to 32 tons per square inch, and not less than 23 per cent. for material required to have a tensile breaking strength of 26 to 30 tons per square inch. For material under $\frac{1}{4}$ in thickness the respective elongations may be 3 per cent. less. Where short test pieces are used for round bars, the elongations on $3\frac{1}{2}$ diameters shall be 25 and 29 per cent. respectively. Rivet bars shall have an elongation of not less than 25 and 31 per cent. on gauge lengths of 8 and $3\frac{1}{2}$ diameters respectively, intermediate proportions of length to diameter to have intermediate percentages.

7. Cold and temper bend test pieces must withstand, without fracture, being doubled over until the internal radius is equal to $1\frac{1}{2}$ times the thickness of the sample and the sides are parallel. Bending tests may be made by pressure or by blows.

8. The stamping of the material, rejection for defects, the issuing of advice notes, &c., must be carried out in the same way as for ship steel, the Rules for which are as follows:—

In the event of any test piece failing to meet the requirements, unless two further tests prove satisfactory, the plate or bar represented by such sample piece must be rejected. Further tests of pieces taken from the same charge shall then be made, and in case of these also failing, the whole of the material produced from such charge is to be rejected.

All material which has satisfactorily passed the requirements must be clearly stamped by the manufacturer, as described in the paragraph beginning at line 6 on this page.

Before the mill sheets are signed, the manufacturer must furnish the Surveyor with a certificate guaranteeing that the material has been made by the "Open Hearth" process, and that it has withstood satisfactorily the tests above described. The following form of certificate will be accepted if printed on each mill sheet with the name of the firm, and initialed by the test house manager:—

> "We hereby certify that the material described below has been made by the 'Open Hearth' process, and is that which has been satisfactorily tested in the presence of the Surveyors in accordance with the tests of the British Corporation Registry."

When it is found necessary from failure in testing, surface defects, or other sufficient cause, to reject any of the material submitted for testing, the Surveyor is to satisfy himself that the stamp of the Corporation upon the rejected material is obliterated by centre punch marks, thus—B·C.

The maker must adopt a system of marking the ingots, billets, slabs, test pieces, and finished material, which will enable all material to be traced to its tests and original charge. The Surveyor must be given every facility for tracing the material to the respective charges from which it is made, and he should be furnished with two copies of advice notes of such material as may have been satisfactorily dealt with, for his signature—one to be forwarded by the manufacturer to the boiler-maker, the other to be retained for the use of the Surveyor in attendance at the boiler works.

Where steel is not produced in the works at which it is rolled, a certificate shall be supplied to the Surveyor deputed to witness the testing of the material, stating the Open Hearth process by which it is made, the name of the steel maker who supplied it, and the numbers of the charges, for reference to the books of the steel maker. The number of the charge shall be marked on each plate or bar for the purpose of identification.

The foregoing tests shall be made at the place of manufacture prior to the despatch of the material; but in the event of any of the material proving unsatisfactory in the process of being worked, it shall be rejected, and such further tests of material from the same charge may be made as the Surveyor in attendance may consider desirable, and material then found defective shall also be rejected, notwithstanding any previous certificate of satisfactory testing.

9. In addition to the tests for rivet steel specified above, the manufactured rivets are to be tested, hot and cold, by bending, crushing under the steam hammer, or in such other way as may be required, a sufficient number of sample rivets being taken indiscriminately for the purpose. The shank must stand being doubled together cold, without fracture, and the rivet head must stand being flattened hot until its diameter is $2\frac{1}{2}$ times the diameter of the shank. The rivets are not to be used until such testing has been carried out to the satisfaction of the Surveyors.

10. All plates which have been welded, or locally heated for furnace work, and plates in which the rivet holes are punched instead of being drilled, must be annealed after this work is carried out. Steel stay bars which have been worked in the fire must be subsequently annealed, and in no case are steel stays to be welded.

11. **Strength Calculations.**—The sizes and arrangement of the different parts for a given working pressure, or the working pressure suitable to a given size and arrangement of material, must be in accordance with the following formulæ and rules. All manhole or similar openings in boilers are to be efficiently compensated:—

12. **Cylindrical Shells of Steel** whose tensile is S tons:—

$$W = \frac{(T-2) \times E \times S}{D \times C}.$$

Where C = 2·75 when the longitudinal seams are fitted with double butt straps of equal width, and of thickness at least equal to that obtained from the formula for butt straps (see par. 13).

C = 2·85 when the double butt straps are of unequal width, *i.e.* one strap not covering the outer row of rivets, with the thickness as before.

C = 2·95 when the longitudinal seams are lap joints.

T = thickness of shell plate, in 32nds of an inch.

E = the least percentage of strength of longitudinal joints, found as follows:—

For the plate at the joint, $E = \frac{p-d}{p} \times 100.$

For the rivets at the joint, $E = \frac{n \times a}{p \times t} \times \frac{r}{s} \times 100.$

Where p = pitch of rivets in outer rows, in inches.

d = diameter of rivet holes, in inches.

n = number of rivets used per pitch.

a = sectional area of one rivet, in square inches.

t = thickness of plate, in inches.

r = 23 for steel and 19 for iron rivets.

s = minimum tensile strength of shell plates, as specified.

Where rivets are in double shear, $a \times 1{\cdot}875$ is to be used instead of a.

D = greatest internal diameter of shell, in inches.

W = working pressure, in lbs. per square inch.

13. **Double Butt Straps.**—The thickness of each plate is to be in accordance with the following formula:—

$$T_1 = T \frac{5(p-d)}{8(p-kd)}.$$

Where T_1 = thickness of outer strap, in 32nds of an inch.
$T_1 + 4$ = thickness of inner strap.
T = thickness of shell plate, in 32nds of an inch.
p = pitch of rivets in outer rows, in inches.
d = diameter of rivet holes, in inches.
k = ratio of pitch of rivets in outer rows to that in centre rows.

14. Flat Surfaces supported by Stays :—

(*a*) $$\text{W.P.} = \frac{C \times (T-1)^2}{P^2 + p^2}\text{; and thickness } T = \sqrt{\frac{\text{W.P.}(P^2+p^2)}{C}} + 1.$$

(*b*) Flat surfaces not exposed to flame, fitted with doublings between the rows of supporting stays, the doublings being at least 0·6 of the thickness of the plates to which they are riveted :—

$$\text{W.P.} = \frac{(T-1)^2 + 0{\cdot}55\,t^2}{P^2+p^2} \times C\text{; and } T = \sqrt{\frac{\text{W.P.}(P^2+p^2)}{C} - 0{\cdot}55\,t^2} + 1.$$

(*c*) Flat surfaces not exposed to flame, strengthened by outside doublings or washers on the stays, and fitted with nuts outside and inside :—

$$\text{W.P.} = \frac{(T-1)^2 + (C \times t)^2 \times 100}{P^2+p^2}\text{; and } T = \sqrt{\frac{\text{W.P.}(P^2+p^2)}{100} - (C \times t)^2} + 1.$$

Where W.P. = working pressure, in lbs. per square inch.
T = thickness of plate, in 32nds of an inch.
t = thickness of doublings or washers, in 32nds of an inch.
P = distance between the rows of stays, in inches.
p = pitch of stays in the rows, in inches.
C = 50 for plates fitted with screwed stays having riveted heads.
C = 75 for plates fitted with screwed stays having nuts outside.
C = 100 for plates fitted with stays having nuts inside and outside.
C = 0·15 for loose washers not less than 0·6 the thickness of plate and three times the diameter of stay.
C = 0·35 when the washers are riveted to plates and in diameter two-thirds pitch.
C = 0·55 when there are doubling strips riveted to plates.
C = 0·85 when there is a doubling plate over the whole surface.

Note.—For front plates in the steam space, which are not protected against the direct action of the flame, the constants given above are to be reduced 20 per cent. Stays which are screwed into plates of less thickness than $\frac{7}{16}$ inch are not to have riveted heads, but are to be fitted with nuts.

15. **Tube Plates with Tubes in Nests:—**

$$\text{W.P.} = \frac{80 \times (\text{T}-1)^2}{\text{P}^2 + p^2}; \text{ and } \text{T} = \sqrt{\frac{\text{W.P.}(\text{P}^2+p^2)}{80}} + 1.$$

T = Thickness of plate in 32nds of an inch.

$$\text{W.P.} = \frac{875 \times (\text{P}-d) \times \text{T}}{\text{L} \times \text{P}}; \text{ and } \text{T} = \frac{\text{W.P.} \times \text{L} \times \text{P}}{(\text{P}-d) \times 875}.$$

P = greatest pitch of stay tubes from centre to centre, in inches.
p = least pitch of stay tubes from centre to centre, in inches.

For wide spaces between nests of tubes see par. 14.

When girders are fitted to the tops of combustion chambers, the thickness of the tube plates is not to be less than given by the preceding formula or than is found from the formula:—

$$\frac{\text{W} \times \text{L} \times \text{P}}{(\text{P}-d) \times 1800} = \text{T}.$$

Where W = working pressure in lbs. per square inch.
L = width of combustion chamber over the plates, in inches.
P = horizontal pitch of tubes, in inches.
d = inside diameter of plain tube, in inches.
T = thickness of tube plate, in sixteenths of an inch.

16. **Stays:—**

$$\sqrt{\frac{\text{S} \times \text{W}}{\text{C}}} + \tfrac{1}{8} = \text{D}.$$

Where S = surface, in square inches, supported by the stay.
W = working pressure in lbs. per square inch.
D = effective diameter of stay, in inches.
C = 8250 for steel screwed stays, and for iron screwed and longitudinal stays made from tested iron bars. Steel stays are to comply with the tests specified above. Iron stays are to be of the best quality and are to have a tensile breaking strength of not less than 20 tons per square inch. Iron bars for stays which require to be tested are to have a tensile strength of not less than 21½ tons per square inch; the elongation on a gauge length of 8 diameters is not to be less than 20 per cent. nor 27 per cent. on a gauge length of 3½ diameters.
C = 6500 for untested iron screwed stays.
C = 9500 for longitudinal steel stays.
C = 7000 for longitudinal iron stays.
C = 5000 for welded iron stays.

17. **Stay Tubes** are not to be subjected to a greater stress than 7500 lbs. per square inch.

18. **Circular Furnaces.**—Thickness of plain furnaces, combustion chamber bottoms, and of furnaces with Adamson rings pitched more than 20 inches apart:—

$$\frac{51{\cdot}5}{D}\Big(18{\cdot}75\,T-(L\times 1{\cdot}03)\Big)=W.$$

Suspension bulb furnaces to be:—

$$W.P.=\frac{510\times(T-1)}{D};\text{ and }T=\frac{W.P.\times D}{510}+1.$$

Corrugated, ribbed, and suspension furnaces (Fox, Purves, Morison, Deighton, and Brown's cambered):—

$$W.P.=\frac{480(T-1)}{D};\text{ and }T=\frac{W.P.\times D}{480}+1.$$

Where T = thickness of plate, in 32nds inch.
L = length of furnace, or combustion chamber bottom between the points of support, in inches.
D = smallest outside diameter of furnace, in inches.
W.P. = working pressure, in lbs. per square inch.

19. **Girders for Combustion Chamber Tops:**—

$$\frac{C\times d^2\times T}{(L-P)\times D\times L}\times\frac{(n+1)^2}{n(n+2)}=W\text{ when }n=2,\ 4,\text{ or }6.$$

$$\frac{C\times d^2\times T}{(L-P)\times D\times L}=W\text{ when }n=1,\ 3,\text{ or }5.$$

Where C = 495 for steel and 450 for iron.
d = depth of girder at the centre, in inches.
T = thickness of girder, in inches.
D = distance from centre to centre of girders, in inches.
L = length from tube plate to tube plate, or from tube plate to back of combustion chamber, in inches.
n = number of stays fitted with each girder.
P = pitch of stays supported by girder, in inches.

20. **General Construction.**—Each boiler must have at least one glass water gauge, two test cocks direct on the boiler shell, or an additional glass water gauge, and one steam pressure gauge. Where the pillar and connections of the water gauge are 2 inches or more in diameter, the test cocks may be fitted thereon. Double-ended boilers to have these fittings at each end. One salinometer cock to each boiler. Where a pillar is attached by pipes to steam and water spaces, cocks should be fitted to the boiler at the ends of these pipes. When water gauge pillars and their boiler connections are less than 2 inches diameter and no cocks on the boiler connections, the passage in the pillar between the gauge cocks is to be closed so as to permit of each end being blown through separately.

21. A stop-valve is to be fitted on each boiler, so that any one of a series of boilers may be worked independently. The neck of the stop-valve to be as short as possible. All boiler and engine stop-valves to be tested to at least twice the W.P.

22 Two safety-valves required for each main boiler, and one for each superheater, and they must be tested under steam and set to a pressure not more than 3 per cent. in excess of the intended W.P. The combined area of the valves is to be sufficient to prevent the steam accumulating to more than 10 per cent. of the W.P., during fifteen minutes full firing with main engines stopped. If the boilers be supplied with forced draught, the valve area must be such that the same conditions may be met.

23. Easing gear is to be so arranged that the safety-valves on any one boiler may be lifted without interfering with those on any other boiler.

24. All boiler blow-off pipes are to have cocks or valves on the hull plating fitted with spigots extending through the plating, and plate flanges round same on the outside; in addition, blow-off cocks or valves are to be fitted to the boiler. (See also sec. 33, par. 12.)

25. Manhole doors in boilers to be not less than 16 ins. by 12 ins.

26. Headers for water tube boilers and superheaters are to be tested by a hydraulic pressure of three times, and other parts by one of double W.P The tubes to be of solid drawn steel and stand being flattened cold until the sides are not further apart than twice the thickness of the tube; they must also withstand expansion cold without cracking or splitting, until the diameter is increased by 10 per cent. in tubes $\frac{3}{16}$ thick or less, and 8 per cent. in tubes of greater thickness. The finished tubes should not exceed 27 tons per sq. in. tensile strength and should show an elongation of 20 per cent. on a length of 8 inches; at least one tube in fifty is to be subjected to flattening and expansion tests and all the tubes to a hydraulic pressure of 1500 lbs. per sq. in. The thicknesses of the tubes in inches are not to be less than given by the formula:—

$$T = \frac{D \times W}{6000} + \frac{1}{16}$$

where D = internal diameter of tube in inches,
and W = working pressure in lbs. per square inch.
T = thickness of tube in inches.

The two rows of tubes which are next the fire in water tube boilers are to be increased at least 10 per cent. in thickness.

Tubes for water tube boilers and superheaters are to be of solid drawn steel. (See present par. 26.)

27. Upon completion, the boilers are to be tested by hydraulic pressure to 1·5 the intended working pressure + 50 lbs., and after being placed in position in the vessel, they must be efficiently secured by brackets and stays to prevent any fore and aft or athwartship movement. Superheaters to be tested to twice the working pressure.

It is strongly recommended, because of the rapid corrosion which takes place in material near them, that the boilers be kept as high as possible above the floors or tank top, and that the under side of the boilers be efficiently insulated. (See sec. 31.)

28. Donkey boilers need not have more than one safety-valve, provided the valve area be not less than half a square inch for each foot of grate surface. In other respects the requirements for donkey boilers are the same as for main boilers.

ENGINES. (Sec. 33.)

1. **Shafting.**—The minimum diameters of crank, thrust, propeller, and intermediate shafts may be found from the following formula, except where the ratio of length of stroke to distance between main bearings is unusual, when they will receive special consideration :—

$$D = \sqrt[3]{\frac{P \times L^2 \times S}{B}} \times C.$$

Where D = diameter of shaft in inches.
P = absolute pressure, *i. e.* boiler pressure + 15 lbs.
S = stroke of engine, in inches.
L = diameter of low-pressure cylinder, in inches.
B = value as given in Table B.
C = 1·0 for crank and thrust shafts.
C = 0·95 for intermediate shafts.
C = for propeller shafts to be taken from the following Table :—

Table C.

Co-efft. of Displ. of vessel at ⅓ moulded depth.	Ratio of Diameter of Propeller to Diameter of Crankshaft.					
	13	14	15	16	17	18
·6	1·0	1·01	1·02	1·03	1·04	1·05
·62	1·01	1·02	1·03	1·04	1·05	1·06
·64	1·02	1·03	1·04	1·05	1·06	1·07
·66	1·03	1·04	1·05	1·06	1·07	1·08
·68	1·04	1·05	1·06	1·07	1·08	1·09
·70	1·05	1·06	1·07	1·08	1·09	1·10
·72	1·06	1·07	1·08	1·09	1·10	1·11
·74	1·07	1·08	1·09	1·10	1·11	1·12
·76	1·08	1·09	1·10	1·11	1·12	1·13
·78	1·09	1·10	1·11	1·12	1·13	1·14
·80	1·10	1·11	1·12	1·13	1·14	1·15

The value of the divisor C in the formula depends on the ratio $\frac{L^2}{H^2}$ where L = diameter of low-pressure cylinder and H of high-pressure cylinder, in inches:—

Table B.

$\frac{L^2}{H^2}$	Two Cranks at 90° Compound or Quadruple, also Three Cranks at 120° Quadruple Expansion.	Three Cranks at 120° Triple Expansion.	Four Cranks at 90° Quadruple Expansion.
Ratio 3	9,910	..	..
,, 3½	10,160	..	..
,, 3¼	10,410	..	..
,, 3⅜	10,660	..	..
,, 3½	10,910	..	..
,, 3⅝	11,160	..	..
,, 3¾	11,410	..	..
,, 3⅞	11,660	..	..
,, 4	11,910	..	..
,, 4⅛	12,160	..	..
,, 4¼	12,410	..	..
,, 4⅜	12,660	..	..
,, 4½	12,910	13,650	..
,, 4⅝	13,375	14,160	..
,, 5	13,840	14,670	..
,, 5½	14,305	15,180	..
,, 5½	14,770	15,690	..
,, 5¾	15,235	16,200	..
,, 6	15,700	16,710	..
,, 6½	16,630	17,730	..
,, 7	17,560	18,630	..
,, 7½	18,410	19,530	..
,, 8	19,260	20,430	22,660
,, 8½	20,110	21,330	23,660
,, 9	20,960	22,200	24,660
,, 9½	21,750	23,070	25,660
,, 10	22,540	23,940	26,580
,, 10½	23,330	24,810	27,500
,, 11	24,120	25,660	28,420
,, 11½	24,900	26,500	29,340
,, 12	25,680	27,340	30,260

Intermediate ratios to have intermediate divisors.

The least diameter of shafts for direct-coupled and geared turbine engines may be obtained from the following formula:—

$$D = \sqrt[3]{\frac{P \times 65}{R}} \times C.$$

Where D = diameter of intermediate shafts.
P = shaft horse power.
C = 1 R = revolutions per minute.
C = 1·05 D for intermediate thrust shafts between collars
C = 1·1 D propeller shafts.

With geared turbines where the revolutions per minute of the propeller are comparatively low, the value of C for the propeller shaft is not to be less than 1·1 D, nor than the values of C given in Table C, substituting the diameter of intermediate shafting for diameter of crank shaft in obtaining the ratio of propeller to shaft.

Thrust and propeller shafts may be gradually tapered off to the diameter of the intermediate shafting. The central hole through hollow shafting may have a diameter equal to one-third that of the intermediate shafting without increasing the diameters of the shafting required by the formula.

2. The webs of built crank shafts are to be keyed as well as shrunk on to the shaft, and the diameter of the shaft should be increased in way of the web to make up the loss of sectional area at the key way; efficient dowel pins should be fitted in the crank pins.

3. Propeller shafts must be forged from selected scrap iron, rolled iron bars, or a single steel ingot. It is recommended that shaft liners be fitted in one length, that the inboard ends of all shaft liners be tapered, and that the space between the after end and the propeller boss be made watertight.

4. **Forgings.**—Shafts, piston rods, connecting rods, and other important working parts of the machinery, are to be made from selected scrap iron or ingot steel in accordance with the following requirements. Other material, such as nickel steel, etc., will be accepted after compliance with such special tests as may be imposed. All shafts, etc., are to be subjected to examination during the progress of manufacture, when rough turned and when finished.

5. Iron forgings are to be in tensile strength between the limits of 20 and 24 tons per square inch, and to show an elongation of 20 per cent. on a gauge length corresponding to $3\frac{1}{2}$ diameters of the test piece. Samples, 1 inch square, are to stand being bent cold to 90° over a radius of $1\frac{1}{2}$ inches without fracture.

6. Steel forgings for machinery are to be forged from the lower two-thirds of ordinary ingots made by the "open hearth" process; the material is to be tested, and the sectional area of the body of the forging, when it leaves the hammer, should not exceed one-fifth, and no part of the forging should be more than three-fourths the area of the original ingot. Forgings are to be properly annealed in an annealing furnace, and must be free from defect. The tensile strength of the forgings is not to exceed 35 tons per square inch, without special sanction, the elongations are to vary from 30 per cent. at 27 tons to 22 per cent. with 35 tons tensile strength, measured on test pieces of which the gauge length is not less than $3\frac{1}{2}$ times their diameter; samples, 1 inch by $\frac{3}{4}$ inch, with edges rounded to $\frac{1}{16}$th radius, must stand being bent cold to 180° without fracture over a radius of $\frac{1}{4}$ inch for material under 32 tons tensile strength, and $\frac{3}{4}$ inch between 32 and 35 tons tensile strength. The test pieces are to be cut lengthwise from a part of the forging which is not of less sectional area than the body of the forging, and they are not to be detached until they have been stamped by the Surveyors, and until the forging

has been annealed, in cases where subjected to the annealing process. One tensile and one bend test are to be taken from each forging, except in the case of forgings over 3 tons weight and of all propeller shafts, where tests will be required from each end of the forging. Where a number of small forgings are made from one ingot, tests representing each end of the ingot will be sufficient. All ingot steel forgings are to be legibly marked in such a way that they can be traced to the charge from which the material has been made, and these which have satisfactorily passed the requirements are to be clearly stamped "B.C.," and with the identification marks furnished by the Surveyor.

7. **Castings.**—Steel castings are to be made in accordance with the requirements of sec. 4, pars. 15–17, when their use is sanctioned for parts of the machinery which are usually forged. Where steel castings are used instead of the iron castings in ordinary use, they are to be subjected to percussive and hammering tests only. Castings for high-pressure cylinders of reciprocating engines are to be tested to 1·5 W.

Castings for cylinders of turbine engines are to be tested under hydraulic pressure after having been rough bored, to the following requirements:—

H.P. cylinders, admission end, are to be tested to 1·33 W
H.P. ,, exhaust end, ,, ,, ,, ,, W
I.P. ,, ,, ,, ,, ,, ,, 1·5 V
L.P. ,, admission end, ,, ,, ,, ,, 1·5 V
L.P. ,, exhaust end, ,, ,, ,, ,, 30 lbs. per sq. inch.

Where W = working pressure of boilers.
V = pressure to which cylinder safety valves are to be adjusted.

The provisions in section 3 are as follows:—

(15) Where steel castings are intended to be used instead of forgings, every care is to be taken to avoid abrupt changes in sectional area, and each casting must be thoroughly annealed and subjected to percussive and hammering tests, as well as to tests for tensile strength and ductility. The castings should be made by the "open hearth" process, and are to be accurately moulded and free from defects. The material is to have a tensile strength of not less than 26 tons and not exceeding 35 tons per sq. in., with a corresponding elongation varying from 20 per cent. with 26 tons to 15 per cent. with 35 tons, measured on test pieces of which the length between the gauge points is not to be less than $3\frac{1}{2}$ times the diameter, and the sectional area not less than $\frac{1}{4}$ sq. inch.

(16) The sample pieces for testing must have formed part of the actual casting submitted for approval, must have been subjected to similar and simultaneous annealing, and are not to be detached from the casting until after the annealing is completed, nor until they have been stamped by the Surveyor. The piece for ductility test should not be less than 1 inch by $\frac{3}{4}$ inch, with edges rounded to a radius of $\frac{1}{16}$ inch, and must stand being bent cold through an angle of 120°, over a radius of 1 inch, without any appearance of fracture.

(17) Castings of propeller frames, posts, single-plate rudder frames, brackets, and quadrants, must be raised to a height of 6 to 10 feet, according to the character and form of the castings, and dropped bodily on to a hard surface, holes being prepared to receive bosses or similar projections. When large propeller frames are cast in one piece, they may be dropped through an angle of 45° instead of being lifted bodily. Castings of a complex design may, at the discretion of the Surveyor, be exempted from the drop test, and where the material is thin the limits of elongation may be reduced by 5 per cent. All castings are to be slung clear of the ground and well hammered all over with a heavy sledge hammer to test the soundness of the material.

8. **General Construction.**—When the engine bed plate or thrust block is fastened direct to the tank, the top plating in way of same must not be less than $\frac{4}{10}$ inch thick and must be increased according to the size and power of the engines, and the double bottom efficiently stiffened under the thrust block, to the approval of the Committee. The holes for the holding-down bolts are to be tapped through the plate, and the bolts properly screwed in, and fitted with lock nuts underneath. The holding-down bolts must be kept as close to the angle bar as possible, and if they pass through the bar the flanges of the bars must be of sufficient breadth to take the nuts.

9. Two bilge and two feed pumps are to be fitted on the main engines in vessels of 180 feet length and upwards, and are to be so arranged that one of either set may be examined while the other is at work, or equivalent independent pumping arrangements are to be provided. In vessels less than 180 feet one bilge and one feed pump or equivalent independent pumps are to be provided; each bilge pump must be adapted to draw from any compartment of the vessel. All feed pumps worked by the main engines are to be fitted with spring-loaded relief valves. A bilge suction to the circulating pump, having a diameter about twice that of the engine room bilge suction, or a bilge injection valve, is to be fitted in each case.

10. The design and arrangement of pumps, valve chests, suction and delivery pipes, and all cocks and sea connections should be such as to prevent the possibility of water being run into the vessel accidentally.

11. All evaporators and high-pressure feed heaters must be satisfactorily tested by hydraulic pressure before leaving the works of the manufacturer. Feed-water filters between pumps and boilers to be tested to 20 per cent. above double working pressure.

12. All sea-cocks and valves connecting suction pipes are to be placed, if practicable, above the level of the stokehold and engine room platforms, and so as to be easily worked from those platforms. Sea-cocks must be fitted direct on to the hull plating and secured with tap bolts or with bolts having countersunk heads. Discharge pipes should be placed above the deep load line, with valves fitted in an accessible position on the vessel's side.

13. Donkey pumps adapted to draw from the sea and hotwell, to

supply the boilers with water through separate auxiliary feed pipes and check valves, to deliver water on deck, and to draw the bilge from each compartment, are to be fitted in all cases. When there are more main boilers than one, and the donkey pump is arranged to draw from the bottom of the boilers, for circulating purposes, the suction pipes are to have non-return valves on the boiler connections, and a shut-off valve or cock on the main pipe from the boilers to the pump.

14. All pipes extended through bunkers, or other compartments beyond the machinery space, must be well protected with strong casings.

15. **Steam and Feed Pipes** are to be of copper, wrought iron, or steel of thicknesses in accordance with the following formulæ. All main steam pipes must have efficient provision for expansion, and are to be so arranged that water cannot lodge in any part of them, or, if this be impracticable, so that they can be properly drained. Auxiliary steam pipes which exceed 4 inches in diameter, and all main and refrigerator machine steam pipes, are to be tested by hydraulic pressure, in the presence of the Surveyor, to at least twice the working pressure when made of copper, and to three times the working pressure when made of iron or steel; and all boiler feed pipes are to be subjected to a hydraulic pressure at least 20 per cent. greater than is required for steam pipes. When pipes have screwed flanges they are to be screwed with vanishing threads, and the thread is not to extend beyond the back of the flange collar.

Thickness of brazed copper pipes in 100ths of an inch $= \frac{W.P. \times D}{45} + 3.$

Thickness of solid-drawn ,, ,, ,, $= \frac{W.P. \times D}{60} + 3.$

$\frac{W.P. \times D}{90} + 12$ = thickness in 100ths of an inch of iron and steel pipes lap-welded; but the minimum thickness for bending or to have screwed ends is $\frac{3}{16}$ inch.

$\frac{W.P. \times D}{120} + 12$ = thickness in 100ths of an inch of solid-drawn steel pipes.

Where D is the internal diameter of pipe in inches.
W.P. is the working pressure for steam pipes.
W.P. = boiler pressure × 1·2 for feed pipes.

16. **Spare gear**, consisting of the following articles, must be supplied to all vessels built to the requirements of the British Corporation:—

2 connecting rod top-end bolts.
2 ,, bottom-end bolts.
2 main-bearing bolts.
1 set of coupling bolts.
1 ,, feed and bilge pump valves.
1 ,, piston springs, if common springs are used.
1 safety-valve spring of each size fitted, but not fewer than 1 spare spring for each 6 safety-valves fitted.
½ set of fire bars for each boiler.

A number of bolts, nuts, and studs of assorted sizes, including at least six cylinder cover bolts or studs, and six valve chest cover bolts or studs.

Bar and plate iron in various sizes.

17. **Completion of Machinery.**—When all the connections are made on board, the safety-valves set, and the machinery completely fitted up, a trial under steam is to be made with the Surveyor in attendance, and if everything is then found to be working satisfactorily the Committee will grant a Certificate, and the machinery will be entered in the Register Book, under the Class of the Hull, thus—M. B. S. *, with date of completion.

STEAM PUMPING ARRANGEMENTS. (Sec. 34.)

1. **Bilge Suctions in Vessels with Ordinary Floors.**—Not less than three steam pump suctions, of the sizes given in the Table, are to be fitted at the after end of each compartment, one being placed at the centre line and one in each wing. At the ends of the vessel, or in compartments where the rise of floor is considerable, one centre line suction will be accepted.

2. **Bilge Suctions in Vessels with Double Bottoms.**—There must be one suction in each wing at the after end of each compartment, of the size given in the Table for wing suctions with two suctions in each hold. Where wells are adopted, the suctions are to be as required for vessels with ordinary floors; but it is recommended that the wing suctions be placed outside the wells, and the openings into the wells fitted with non-return valves.

3. **Double Bottoms.**—At least one centre suction, of the size required by the Table for the size of the tank, is to be fitted at the after end of each compartment in the double bottom. In vessels where the rise of floor is less than 1 inch per foot, and the depth to the top of the midship erections is 75 per cent. or more of the vessel's breadth, centre and wing suctions should be fitted in the midship tanks.

4. **Peaks and Deep Tanks.**—Where peaks or cargo spaces are used as ballast tanks, suctions of the size required by the Table for the size of tank are to be fitted. Cargo spaces used as ballast tanks are to have satisfactory arrangements for cutting off the bilge suctions when used for water and the ballast suctions when used for cargo; also for draining the tops of the tanks.

5. **Tunnel Wells** must in all cases be fitted with suctions, of not less size than is required for wing suctions with three suctions in each hold.

6. **The Main and Donkey Pumps** are to draw from all compartments, and in addition the donkey is to have a separate bilge suction in the engine room. The pipes connecting the bilge pumps with the valve chests are not to be of less diameter than required for centre suctions, and the ballast pump connections of not less diameter than required for a single suction to the largest tank suction in the vessel. The pumps must be of sufficient capacity to give a speed of water

through the pipes of not less than 400 feet per minute, under ordinary working.

7. **Bilge and Ballast Suction Pipes** are to be efficiently secured, and straps are to be fitted at the middle of the length of each range of pipes to prevent fore and aft movement. Efficient expansion joints are to be fitted, and where the connections at the ends of each range of pipes are made with lead bends, the radii of the bends and the distance between the centres of the radii should each be equal to three diameters, and the length of the bend to eight diameters of the pipe.

8. **All Roses and Boxes** to be placed so as to be easily accessible for examination and cleaning. It is recommended that each engine-room bilge suction should have a mud box placed above the level of the floor plates, with a tail pipe led from it into the bilges, and without the usual rose box in the bilges. Vessels under 180 feet in length should have provision made on the donkey pump for the attachment of a flexible hose, which should be supplied in addition to the engine-room requirements.

9. **Sounding Pipes** are to be fitted to each compartment and ballast tank, with a small doubling plate bedded in the cement under each pipe, for the rod to strike on. These pipes must be fitted, without bends, directly into the compartment intended to be sounded, and should in all holes extend to the upper deck.

10. **Air Pipes**, not less than 2 inches in diameter, are to be fitted at each corner of each ballast tank; this requirement may be modified in the case of small tanks and increased for large tanks. Efficient arrangements must be made to permit of the air getting freely to the pipes while the tanks are being filled.

Inside Diameters of Pipes (Minimum Requirements).

Bilge Suctions.			
Length of Vessel.	Centre and all Engine Room Suctions.	Wing Suctions with Two Suctions in hold.	Wing Suctions with Three Suctions in hold.
	Inches.	Inches.	Inches.
Under 180 feet, . . .	2	...	...
180 feet and under 230 feet, .	2¼	2	2
230 ,, ,, 280 ,, .	2½	2¼	2
280 ,, ,, 330 ,, .	2¾	2½	2¼
330 ,, ,, 390 ,, .	3	2¾	2½
390 ,, ,, 450 ,, .	3¼	3	2½
450 ,, ,, 520 ,, .	3½	3¼	2¾

Inside Diameters of Pipes (Minimum Requirements)—*contd.*

TANK SUCTIONS.

CAPACITY OF TANK.	Diameter of Suction.	CAPACITY OF TANK.	Diameter of Suction.
	Inches.		Inches.
Under 20 tons, . .	2½	270 tons and under 365 tons,	5
20 tons and under 40 tons,	2¾	365 ,, ,, 480 ,,	5½
40 ,, ,, 60 ,,	3	480 ,, ,, 625 ,,	6
60 ,, ,, 85 ,,	3¼	625 ,, ,, 800 ,,	6½
85 ,, ,, 120 ,,	3½	800 ,, ,, 1000 ,,	7
120 ,, ,, 190 ,,	4	1000 ,, ,, 1300 ,,	7½
190 ,, ,, 270 ,,	4½		

ELECTRIC LIGHTING. (Sec. 35.)

While the Committee do not consider it advisable at the present time to formulate definite rules for fitting vessels with electric-light installation, they are prepared to approve of such installations as may be carried out on well-considered and safe systems. It is suggested, therefore, that a complete specification of the apparatus and method of wiring proposed should be submitted for the consideration of the Committee, together with samples of the wires, switches, cut-outs, etc., if required.

In drawing up the specification, the following points should be kept in view:—

1. Dynamos, motors, main and branch cables should be so placed that the compasses will not be injuriously affected by the electric current. Tests should be made while the compasses are being adjusted, in order to prove that this condition has been satisfactorily fulfilled.

2. Conductors, switches, cut-outs, and hull connections, if any, should be so arranged as to be always and easily accessible.

3. Switches and cut-outs should be on non-inflammable bases—preferably of porcelain—and the switches should be of the quick-break pattern.

4. Cut-outs should be fitted in every case at each reduction in size of wire in the single wire system, and on both wires where double wiring is adopted. It is recommended that tin wire cut-outs of standard sizes be used, the number of sizes being kept as small as possible.

5. Hull connections for "single wire" systems should be made with brass screws of large surface, carefully fitted.

6. Conductors should be made of the best copper, and the insulation resistance should not be less than 600 megohms per mile—a certificate from the manufacturer to this effect to be furnished if desired. Vulcanized rubber insulation is strongly recommended, but other materials may be adopted if possessing equal insulation resistance and durability.

No single wire should be smaller in diameter than No. 18, or larger than No. 16 standard wire gauge; where sizes above No. 16 are required the conductors should be formed of small wires stranded together.

The current density in the conductors should not exceed 1000 amperes per square inch.

All conductors should be protected by efficient casings, or lead covered, or armoured, and the covers on wood casings should be portable and fastened with screws. Conductors which are exposed to excessive heat or damp, or which are led through bunkers or cargo spaces, should be lead-sheathed, and in engine room, stokehold, and tunnel they should be further protected by armour.

Special care should be taken to protect conductors from damp and chafe where they pass through deck or bulkheads.

7. Joints in main cables should be avoided wherever possible, and, when absolutely necessary, they should be so made that the watertightness and insulation resistance of the cables is not affected by them.

Ordinary joints in branches should be properly soldered and thoroughly insulated, resin only being used as the flux for soldering. It is strongly recommended that such a system of distributing and auxiliary switch boards be adopted as will render all ordinary joints unnecessary.

8. A voltmeter should be fitted in every case, and an ammeter where there is more than one dynamo.

9. A final test should be made when the installation is complete, when the insulation resistance over the whole or any part of the system should not be below the following:—

Installations of 25 lights,	.	.	500,000 ohms.
,, 50 ,,	.	.	250,000 ,,
,, 100 ,,	.	.	125,000 ,,
,, 500 ,,	.	.	25,000 ,,
,, 1000 ,,	.	.	12,500 ,,

For intermediate numbers of lights the insulation resistance should be correspondingly proportionate.

For alternating currents the minimum insulation resistance should be twice that given in the Table.

PERIODICAL SURVEY OF ENGINES AND BOILERS.

The engines and boilers of all vessels classed with the Corporation will be required to undergo Special Periodical Surveys at the same times as the Special Surveys on the hull. In cases of accident involving considerable repair and an extensive examination of the

machinery, such examination may, with the sanction of the Committee, be considered equivalent to a Special Survey.

At each Special Survey the cylinders or turbine, pistons, valves, pumps, evaporators, thrust block and condenser, main and tunnel bearings, shafting, and the steam-steering gear must be opened up for examination, and such other parts of the machinery as may be considered necessary are to be examined.

The arrangements for pumping from the several holds, as well as from the engine and boiler space, are to be inspected, and while the vessel is in dry dock, all openings to the sea, together with the cocks and valves in connection with the same, examined; in addition, all iron and steel fastenings of sea-cocks and valves to the shell plating should be removed for examination at each Special Survey No. 3. The propeller shaft should be drawn at least once every two years, and more frequently if considered necessary by the Surveyor, but when liners are fitted solid in one length the shaft need only be drawn once every three years; the Committee are, however, prepared to consider representation from owners as to special circumstances which may modify these requirements in particular cases.

When the after bearing is worn down $\frac{1}{4}$ inch with shafts not exceeding 9 inches in diameter, $\frac{5}{16}$ when over 9 and not exceeding 12 inches, and $\frac{3}{8}$ with shafts over 12 inches in diameter, the bearing must be rebushed.

At each Special Survey, the boilers, superheaters, and safety-valves are to be carefully examined inside and outside, and the safety-valves set to the working pressure, and the main steam pipes should be tested by hydraulic pressure every four years in case of brazed copper pipes, and every six years in case of solid drawn copper pipes and of iron and steel pipes; copper pipes should be annealed before being tested. If at any of these surveys the Surveyor considers it desirable, the actual thickness of plates and strength of stays are to be ascertained in order to determine the future working pressure, and the boilers and superheaters tested by hydraulic pressure.

In addition to the requirements of the Special Periodical Surveys, when six years old, and annually thereafter, the boilers, superheaters, and safety-valves must be carefully examined and the valves set to the working pressure.

If it be found desirable, upon inspection, that any part of the engines or boilers should be examined again within a short period, it will be necessary for the owner to comply with the Committee's requirements in this respect.

The donkey boilers of Sailing Vessels are to be subject to Special Periodical and Annual Survey in accordance with the foregoing requirements.

APPENDIX B.

BUREAU VERITAS RULES AS IN OPERATION ON THE CONTINENT.

(In the United Kingdom this Registration Society accepts and uses the Rules of the Board of Trade.)

For Machinery.*

Classification.—Mechanically propelled vessels will only be granted a class after the machinery has been examined and certified by an Engineer Surveyor of the Bureau Veritas as being in accordance with the rules.

The Special Survey mark ✠ will only be granted in the Register when the boilers and machinery have been constructed under the special inspection of an Engineer Surveyor of the Bureau Veritas.

The above provisions apply also to sailing vessels fitted with auxiliary machinery.

To obtain Special Survey of machinery and boilers the following provisions must be complied with :—

1. Make application in writing to District Surveyor.

2. Submit plans of boilers and safety-valves, and give full particulars of materials to be used.

3. Give Surveyors every opportunity and facility for making full inspections of all parts during manufacture and fixing.

Hand to Surveyors duplicates of all orders for material, and see that they have facilities for testing all material.

4. Notify Surveyors where hydraulic tests as described below can be witnessed :—

Cylinders of Compound, Triple, and Quadruple engines,—H.P. cylinders to be tested to boiler pressure plus 85 lbs. per sq. in. ; L.P. cylinders to 43 lbs. per sq. in. ; I.P. cylinders of triples to 0·8 boiler pressure ; first intermediate cylinders of quadruples to boiler pressure ; and second intermediate cylinders of quadruples to half boiler pressure.

H.P. turbine casings to be tested to one and a quarter times boiler pressure ; L.P. turbine casings to a quarter boiler pressure at admission ends and 28·5 lbs. per sq. in. at exhaust ends ; astern turbine casings to one and a quarter boiler pressure H.P. and 43 lbs. L.P. end.

* These rules are summarised, and not given word for word.

5. Condensers to be tested to 28·5 lbs. per sq. in.

6. Boilers to be tested to twice working pressure up to 142 lbs. per sq. in., but for all higher working pressures the test pressure is to be working pressure plus 142 lbs. per sq. in.

7. Steam and feed pipes to be tested to twice working pressure, and other steam pipes to twice maximum pressure to which they will be subject. These tests may be made after fitting on board.

9. Boilers, after being fitted on board, to be tested again to once and a half working pressure up to 85 lbs. per sq. in., but for all higher working pressures to working pressure plus 85 lbs. per sq. in.

11. The Surveyor will afterwards attend a steam trial and adjust the safety-valves, and will also proceed with the vessel on her trial trip and satisfy himself as to the efficient working of the whole of the machinery.

Maintenance of Class: Annual and Periodical Surveys.—Periodical Surveys of machinery must take place every four years for vessels of the first division, and every three years for vessels of the second and third divisions.

Boilers over twelve years old must undergo the periodical Survey with the corresponding tests every two years. In countries where the testing of boilers is regulated by law, the Surveyors are authorised to conform to the legal requirements.

Propellers must be removed and propeller shafts drawn for examination at least every two years.

Surveyor to be notified whenever boiler, propeller, or tail shaft is removed; also all repairs to be carried out to satisfaction of Surveyor and reported by him to the Administration.

A Survey is to be held in all cases of accident to machinery, and the master of the vessel must notify Surveyor; failing such notification, class may be withdrawn.

An annual Survey will be mainly external so far as main and auxiliary engines are concerned, though the Surveyor is empowered to open up any part that he may think necessary. As regards boilers, the annual Survey will be both external and internal, and the Surveyor will see boilers under steam and readjust safety-valves if necessary.

At periodical Surveys the Surveyor will require to have cylinders opened and pistons removed, and slide-valves and main bearings opened up; and he will also carefully examine condensers, pumps, pipes, cocks and valves, propeller and other shafting and propeller. He will also make a complete examination and hydraulic test of boilers, and may order repairs, lowering of working pressure, or may even condemn boilers. The hydraulic test to be to one and a half times working pressure, as described above.

When boilers twelve years old are being examined the Surveyor will probably require to drill various parts to ascertain thicknesses, and he may require the partial removal of lagging from boilers and steam pipes.

On completion of a periodical Survey, the Surveyor will attend a steam trial and satisfy himself as to the efficiency of the whole of the machinery.

Construction and Arrangement of Engines.—All machinery must be so securely fixed that the motions of the vessel will not cause anything to come adrift; sufficient space must be provided around and above engines and in tunnels, and sufficient ventilation given to enable the engineers to give proper attention whilst engines are running, and also to effect necessary examinations and repairs. Guard plates and handrails must be fitted where required.

Turning gear must, and steam reversing gear should, be fitted to all engines over 500 horse-power.

Condensers are to be of strong construction and fitted with suitable manholes and handholes for examination and cleaning.

Valve-chests, cocks and pipe connections must be so arranged (except ash cocks and water service cocks) that sea water cannot be run into the ship.

All discharge valves to be fitted direct on vessel's skin plating above load line if possible, in accessible positions, and should be capable of being closed in harbour.

All pumping pipes should be easily accessible, and none should be carried through the bunkers without proper protection.

Bilge pumps and donkey engines to pump from all compartments except fore and after peaks; and if there be no well, in holds with double bottoms, a suction must be fitted in each wing gutter. In addition to the suctions from the common valve boxes the engine bilge pumps must be fitted with a direct suction from the engine-room bilges.

When the circulating pump is fitted with a bilge suction, care must be taken to arrange so that sea-water cannot run into the ship; and suitable gratings must be fitted at inlets.

Steam pipes must be so fitted as to allow of expansion; drain cocks must be fitted where necessary; and steam pipes on deck must be protected by casings.

Satisfactory means of communication between engine room and bridge must be fitted.

In vessels under the French flag all pipes must be painted with colours indicating their purposes or uses.

Construction and Arrangement of Boilers.—Boilers must be so secured that neither the rolling of the vessel nor the shock of collision will move them: they must be arranged in spaces of sufficient size for easy working and repair, and the said spaces must be properly ventilated, must have two easily accessible exits, and must have the overhead openings fitted with gratings and shutters or covers.

When donkey boilers and auxiliary machinery are placed outside the main engine and boiler rooms, the spaces provided must be enclosed by steel bulkheads and the decks beneath must be plated and caulked; the spaces must also be properly ventilated.

In all steam ships provision must be made for preventing coal or ashes getting down through stokehole flooring; and in those of more than 800 horse-power a steam ash-hoisting apparatus must be fitted.

Coal bunkers must be kept at least one foot away from boilers or chimney unless insulating material is fitted; and proper ventilating

arrangements must be fitted, especially in vessels with more than one deck. Steam pipes must not pass through bunkers, and if water pipes or electric leads do pass through they must be protected by strong casings.

Oil fuel may be carried in the double-bottom, bunkers, peaks, or other suitable compartments if various conditions, laid down in detail in the Rules, are complied with. If the oil fuel has a lower flash-point than 150° F., the proposed arrangements must be specially submitted to the Administration for approval.

The water-level should, ordinarily, be at least six inches above the tops of the combustion chambers, but in small boilers this may be reduced to four inches.

The minimum level must be distinctly marked as near as possible to the water-gauges.

At least two spring safety-valves of approved design and of area given by following formula must be fitted direct to each main boiler; and they must be provided with easing gear workable both from stokehole and deck, and also with drain-cocks.

When A is area of valve in square inches, that must be provided per square foot of grate surface, and P is the boiler pressure in lbs. per square inch,

$$A = \frac{8{\cdot}7}{\sqrt[4]{(P-18)^3}}.$$

Where forced draught is fitted, the areas of safety-valves must be increased in proportion to the increased evaporative power of the boilers.

For boilers with grate surface not exceeding five square feet only one valve will be required, but no valve of less than 1¼ inches diameter will be accepted. All single-seated safety-valves must be capable of lifting at least one-quarter of the diameter of the valve.

All steam receptacles receiving from the boilers of more than 3·5 cubic feet capacity must have a safety-valve when the effective pressure is over 4·27 lbs.

Each boiler (except donkey boilers, etc.) shall be supplied by two separate feed pumps (one at least independent of main engines), pipes, and check-valves, each of sufficient size to feed the boiler in all circumstances.

Every single-ended boiler must be fitted with two water-level indicators, one at least being a glass water-gauge; the second may consist of a set of test cocks (ordinarily three in number, but for small boilers two). Double-ended boilers must have a water-gauge at each end. When test cocks are fitted for pressures over 114 lbs. per square inch, long rods or handles for working them must be supplied.

A stop valve or cock must be fitted direct to the boiler shells at every outlet for steam, and the main stop valves should, where possible, be arranged to be worked from deck.

Every cylindrical boiler should be fitted with both a surface and a bottom blow-off valve or cock attached direct to the shell; but for water-tube boilers a bottom blow-off only will be required. Blow-off cocks with spigots and brass or galvanised iron outside washers must also be fitted on the skin plating of the vessel. Arrangements should

be made to ensure that the opening and closing of blow-off cocks can be properly regulated; and where one pipe connects several boilers, cocks or valves for preventing the passage of water from one boiler to another must be fitted.

Every single-ended boiler must be fitted with a pressure-gauge marked to show working pressure, and every double-ended boiler must be fitted with two such gauges. Gauges to be placed in view of firemen, to have cocks for shutting them off from the boilers, and to have lamps fitted near them.

Each boiler must be furnished with a fitting terminating in a flange 1¼ inch diameter and ¼ inch thick for attaching a standard gauge, and also with a suitable connection for making the hydraulic test.

Every boiler must be provided with manholes, mudholes, and sight holes, conveniently arranged for inspection and cleaning. No manhole to be less than 12 inches by 16 inches, and all are to be fitted with compensating rings.

Cast-iron must not be used for manhole doors.

Boiler Shells and Stays.

Circular Shells and Steam-holders with Internal Pressure.

A riveted joint may fail through the tearing of the plate or butt-strap between the rivets, the shearing of all the rivets, or by a combination of the two. The following formulæ apply to these several cases. The plate thickness and the diameter of rivets to be applied to have the highest values which each formula would give separately.

I. *Rupture through Plate.*

The formulæ for working pressure and plate thicknesses are in this case:—

$$\left.\begin{aligned} P &= \frac{2\,a\,R\,(t-0{\cdot}04)}{D} \\ \text{and} \qquad t &= \frac{P\,D}{2\,a\,R} + 0{\cdot}04 \text{ inch} \end{aligned}\right\} \quad . \quad . \quad . \quad . \quad . \quad (I.)$$

Where P = allowed working pressure, above atmosphere, in pounds per square inch.

D = greatest inside diameter of boiler shell, or steam-holder in inches.

t = thickness of shell plates in inches. $t - 0{\cdot}04$ inch represents the thickness left after a reduction of 0·04 inch through corrosion.

R = the tensile stress, in pounds per square inch, which will be allowed in the plate. The value of R will be the breaking strength divided by 4, the latter figure representing the factor of safety for the plate after it has been corroded away by 0·04 inch.

If the actual breaking strength happens to be known by tests carried out to the Administration's satisfaction, it may be applied for finding R; but when, as usual, it is not known, the value of R will be :—

For Steel: the 4th part of the lower limit of tensile strength chosen by the designer, which in such case is to be stated when a boiler design is submitted for approval.

The table annexed shows the values of 2 R for various tensile strengths.

a = ratio of the resistance of the plate left between the holes, to that of the full plates. It will be determined from the following expression :—

$$a = \frac{p - d}{p}.$$

Where p = pitch of rivets in outer row, in inches (see figs. 1 and 2).
d = diameter of rivet holes, in inches.

TABLE SHOWING THE VALUES OF 2 R. = 1120 × TENSILE.

In Formulæ (I.) *and* (IV.) *for various Tensile Strengths of the Material.*

Tensile Strength of Plates in tons per sq. inch.	Value of 2 R.	Tensile Strength of Plates in tons per sq. inch.	Value of 2 R.
32	35,800	25·5	28,600
31·5	35,300	25·0	28,000
31	34,700	24·5	27,400
30·5	34,200	24·0	26,900
30	33,600	23·5	26,300
29·5	33,000	23·0	25,800
29	32,500	22·5	25,200
28·5	31,900	22·0	24,600
28	31,400	21·5	24,100
27·5	30,800	21·0	23,500
27	30,200	20·5	22,900
26·5	29,700		
26	29,100		

N.B.—Boiler designs submitted for approval of Administration must state lowest tensile of steel for various parts and also shearing strength of rivets; if these are not stated, minimum tensile of all plates (including furnace) and stays will be taken as 26 tons, and steel rivets will be understood to comply with Administration's requirements.

In every case copy of specification of material, with name of makers, must be supplied to Surveyor through whom designs are submitted for approval.

II. *Rupture through Rivets.*

In this case the following are the formulæ for finding the allowed working pressure, or required rivet section :—

$$P = \frac{2\,A\,S}{D\,l}$$

and

$$A = \frac{P\,D\,l}{2\,S} \qquad \text{. . . . . (II.)}$$

Where P and D have the same meaning as before, and

l = the length, in inches, of the identical parts into which a riveted joint can be subdivided. In most cases l is the pitch of the rivets in the outer rows (figs. 1 and 2).

In general it depends upon the system of joint adopted.

S = the maximum shearing stress, in pounds per square inch, which will be allowed on the rivets. It will be the 4th part of the actual shearing resistance of the material, which should always be ascertained by tests, when possible. If the actual shearing resistance of the rivet bars is not known, it will be assumed to amount to 0·8 of their tensile strength, and the value of s will be one-fifth part of the lower tensile limit adopted by the designer.

A = the total shearing surface, in square inches, of the rivets (that is, twice the area of the rivet hole when a rivet is in double shear).

Only $\frac{1}{4}$ of the full area are to be taken when the riveting is done by hand.

III. *Combined Rupture through Plate and Rivets.*

This case is only to be examined when the outer row has a wider pitch than the inner ones.

The formula to be applied in this case is :—

$$P = \frac{2(B \times R) + (C \times S)}{D \times l} \qquad \text{. . . . . (III.)}$$

Where P R S D and l have the same meaning as before.

B = the sectional area, in square inches, of the plate on the portion l of the joint along the line of its supposed rupture, assuming that in case the plate is liable to corrosion, its thickness has been reduced by 0·04 inch.

C = the total area of the rivets which are supposed to shear on the length l, corrected, if required, in the same way as prescribed above.

For a rivet in double shear the resistance will be considered as being twice that of one in single shear.

IV. *Rupture through Butt Straps.*

Rupture may take place along one of the inner rows of rivets (see P Q, figs. 1 and 2). The formulæ for this case, based on the same principle as (I.), are:—

$$\left.\begin{aligned} P &= \frac{2\,a\,R(t-0{\cdot}04)}{D} \\ \text{and} \quad t &= \frac{P \times D}{2\,a\,R} + 0{\cdot}04 \text{ inch} \end{aligned}\right\} \quad . \quad . \quad . \quad \text{(IV.)}$$

when plates are exposed to the direct action of products of combustion as in superheaters.

$$P = \frac{1{\cdot}6\,a\,R(t-0{\cdot}19)}{D},$$

when they are protected from direct action.

$$P = \frac{1{\cdot}8\,a\,R(t-0{\cdot}125)}{D},$$

where P D and R have the same meaning as before.

t = thickness in inches, of butt strap, or sum of thicknesses, if there are two straps. (The thickness, of course, not to be less than required for caulking.)

$$a = \frac{q-d}{q},$$

where q = pitch of rivets in the inner row, in inches.

d = diameter in inches of the rivet holes in the inner row.

V. *Combined Rupture through Butt Straps and Rivets.*

Formula (III.) applies to this case, B being the section of the butt strap or straps along which rupture would take place.

Remarks.—No rivet holes to be nearer the edge of any plate than the diameter of the rivet.

In zig-zag riveting, the distance between the rows is to be such that no rupture through plate or butt strap is to be feared along the zig-zag line.

When stays are bolted through the shell, they should be so arranged that they do not weaken the shell plates more than the riveted joints. If the resistance at the stay bolts is the smaller of the two, the plate's thickness shall be determined by it. It will be found from a formula the same as (I.), p and d applying to the stay bolts.

FLAT PLATES.

The allowed working pressure or the thickness of flat plates is to be determined by the following formulæ:—

$$P = \frac{(t-1)^2}{a^2+b^2} \times \frac{T}{C},$$

and

$$t = 1 + \sqrt{(a^2+b^2)\frac{PC}{T}}.$$

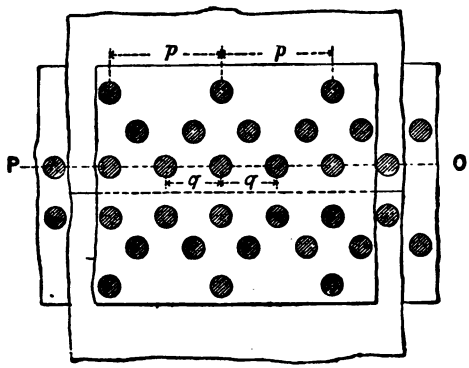


FIG. 1.

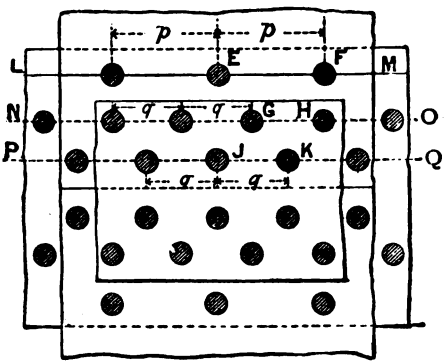


FIG. 2.

Where P = allowed working pressure above atmosphere, in lbs. per square inch.

t = thickness of plate in sixteenths of an inch.

a = pitch of stays, in inches, in one row.

b = distance, in inches, between two rows of stays.

Note.—When plates are effectively stiffened by doubling plates, well riveted thereto, and having a thickness t_1 in sixteenths, the value $(t + \frac{t_1}{2})$ may be substituted for t in the formula.

In case of irregular staying, such as in the annexed sketch (fig. 3),

$$\tfrac{1}{4}(P_1 + P_2)^2$$

shall be taken instead of $a^2 + b^2$.

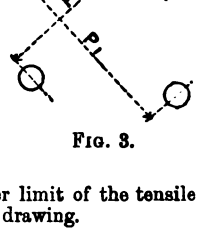


FIG. 3.

T = tensile strength of the material in tons per sq. inch of the original section.

It is to be determined in the same way as for shell plates, that is :—

For steel it will be equal to the lower limit of the tensile stress, which is to be stated on the drawing.

C = a constant, the value of which depends upon the mode of staying as follows :—

C = 0·104 when the stays are screwed into the plates, riveted over, and protected from hot gases; otherwise C = 0·111.

C = 0·079 when the stays are screwed into the plate and fitted with outside nuts at both ends, and protected from hot gases; otherwise C = 0·084.

C = 0·062 when the stays are fitted with inside and outside nuts and washers, provided the diameter of the outside washers be at least 0·4 of the distance between the rows of stays, and the thickness at least $\frac{3}{8}$ that of the plate.

C = 0·555 when the stays are fitted with inside and outside nuts and washers, the outside washer being riveted to the plate and having $\frac{3}{4}$ of the plate's thickness and a diameter equal to 0·6 of the distance between the rows of stays.

C = 0·050 when the outside washers are replaced by strips of plate having a width of at least 0·6 of the distance between the rows of stays with a thickness not less than $\frac{3}{4}$ that of the plate; the strips being well riveted to the plate.

For the values of $\frac{T}{C}$ see following Table.

VALUES OF $\frac{T}{C}$ IN THE FORMULA FOR FLAT PLATES.

Tensile Strength of Plate in tons per square inch.	C=0·111.	C=0·104.	C=0·084.	C=0·079.*	C=0·062.	C=0·055.	C=0·050.
22	198·2	211·5	261·8	278·5	354·8	400·0	440·0
23	201·8	221·1	273·7	291·1	371·0	418·2	460·0
24	216·4	230·8	285·6	303·8	387·1	436·4	480·0
25	225·5	240·4	297·5	316·4	403·2	454·5	500·0
26	234·6	250·0	309·4	329·1	419·4	472·7	520·0
27	243·7	259·6	321·3	341·8	435·5	490·9	540·0
28	252·8	269·2	333·2	354·4	451·6	509·1	560·0
29	261·9	278·8	345·1	367·1	467·7	527·3	580·0
30	270·0	288·5	357·0	379·7	483·9	545·5	600·0

* When plates are not subjected to flame or hot gases C may be reduced from 0·079 to 0·066; or the values of $\frac{T}{C}$ in this column may be multiplied by 1·2.

When the plates are in contact with steam on one side and flame or hot gases on the other, the thickness is to be increased.

For instance, when in return tube boilers, the top front plates are in no way protected from the hot gases, the working pressure or thickness will, in such a case, be determined by the formulæ :—

$$P = \frac{(t-2)^2}{a^2+b^2} \times \frac{0{\cdot}9\ T}{C}.$$

and

$$t = 2 + \sqrt{(a^2+b^2)\frac{P\ C}{0{\cdot}9\ T}}.$$

When the said front plates are protected by a flame plate, no increase of thickness will be required.

When front plates are in two pieces, the lap should be double riveted if the thicker plate is ½ in. or above.

STAYS.

The diameter of stays supporting flat surfaces is to be determined by the following formula :—

$$d = \tfrac{1}{4} \text{ inch} + \sqrt{\frac{Q}{300\ T}}.$$

Where d = effective diameter in inches (for instance, the diameter at bottom of thread in screw stays).

Q = total load on stay in lbs.

T = tensile strength of the material, in tons per square inch.

For steel this tensile strength will be the lower limit chosen by the boiler designer; the actual strength may be applied if it is known from tests.

If the stays are not round, their cross section must be such that the stress per square inch, caused by the load Q, nowhere exceeds one 5·75th part of the tensile strength, after deducting $\frac{1}{16}$ of an inch all round as an allowance for corrosion or wear.

Welding of stays is not allowed.

For high working pressures, such as used in triple-expansion engines, it is recommended to screw all stays into the plates they support, in addition to fitting them with nuts.

This also applies to stay tubes, with the exception that it is recommended that nuts should not be fitted in combustion chambers.

Screwed Stays.

The diameter of fire-box stays is to be determined by the following formulæ :—

$$d = 0{\cdot}177 + \sqrt{\frac{Q}{300\ T}}.$$

For margin stays the formula will be :—

$$d = 0{\cdot}177 + \sqrt{\frac{Q}{260\ T}},$$

d, Q, and T having the same meanings as above.

Vertical Boilers.

When the dished ends in vertical boilers are portions of spheres, their thickness must not be less than—

$$xt = \frac{Pr}{840} + 2.$$

When dished ends in steam chests are portions of spheres,

$$xt = \frac{Pr}{1140} + 3,$$

where t = thickness of plate in sixteenths of an inch ;

x = percentage of joint ;

P = working pressure in lbs. per square inch ;

r = radius of curvature in inches.

When the end is made in one plate, $x = 1$.

When the tops of furnaces are portions of spheres, the thickness must not be less than—

$$xt = \frac{Pr}{600} + 2,$$

where x, t, P, r have the same meaning as before.

Furnaces.

Plain Cylindrical Furnaces.

When plain furnaces are made as truly circular as practicable, and of steel having a tensile strength of not less than 26 tons per square inch, the working pressure P and thickness of plate T may be calculated by the following formulæ :—

$$P = \frac{16{,}000T - 60L}{D},$$

$$T = \frac{P \times D + 60L}{16{,}000},$$

Where T = thickness of plate in inches.
P = working pressure in lbs. per square inch.
D = external diameter in inches.
L = length in inches.

N.B.—When the furnace is in one length L is measured from the centre of rivets at furnace mouth to those connecting back end to the tube plate, or to the commencement of flanging where back end of furnace is flanged. When the furnace is divided into two or more parts by Adamson joints L is measured to the centre of the joint.

For iron and low tensile steel use 14,400.

Furnace plates should not exceed $\frac{15}{16}$ inches in thickness.

Corrugated and Ribbed Furnaces.

The plate thickness is to be found by the following formulæ :—

1. For corrugated furnaces :—

$$T = \frac{PD}{1260} + 2.$$

Where P = working pressure in lbs. per square inch.
T = thickness of plate in sixteenths of an inch.
D = outside diameter in inches measured on the top of the corrugations.

The formula applies to corrugations 6 inches long and $1\frac{1}{2}$ inch deep.

2. For ribbed furnaces, when manufactured to the satisfaction of the Administration :—

$$T = \frac{PD}{1160} + 2.$$

Where P and T are as above.

D = greatest outside diameter between the ribs, in inches.

The formula applies to ribs spaced 9 inches and projecting $1\frac{3}{4}$-inch, the difference between the greatest and the smallest diameters in any part of the furnace not exceeding $\frac{5}{16}$.

3. For bulb furnaces :—

$$T = \frac{PD}{1260} + 2.$$

Where D is the outside diameter in inches measured between the bulbs.

The co-efficients in the above formulæ apply to the case where the tensile strength of the material is 26 tons per sq. in. or above. When it is below 26 tons, the co-efficient is to be reduced $\frac{1}{10}$th for each ton below 26.

When the combustion chamber bottom is cylindrical thickness,

$$T = \frac{P \times 2\,R + 60\,L}{14{,}400}.$$

Combustion Chamber Girders.

The strength of girders on the tops of combustion chambers shall be determined as follows :—

$$P = \frac{C d^2 t}{(W - p)DL}.$$

Where P = working pressure in lbs. per sq. inch.
C = a constant, found as under.
N = number of bolts in each girder.
d = depth of girder, in inches.
t = thickness of girder, in inches.
p = pitch of bolts in girder, in inches.
L = length of girder between supports, in inches.
D = distance between girders, centre to centre, in inches.
W = width of firebox from tube plate to back plate, in inches.

$$C = \frac{13{,}200N}{N+1} \text{ for odd numbers of bolts.}$$

$$C = \frac{13{,}200(N+1)}{N+2} \text{ for even numbers of bolts.}$$

Construction of Engines.

Shafts for Screw Steamers.

A. Crank Shafts.

When the crank of a screw engine is not overhung, the diameter of the shaft shall be determined by one of the following formulæ :—

For non-compound condensing engines :

$$d = \sqrt[3]{\frac{n P L D^2}{C}} \quad \ldots\ldots\ldots \quad (A)$$

For double, triple, and quadruple expansion engines :

$$d = \sqrt[3]{\frac{P L (n_1 D_1^2 + 0.1\,n\,D^2)}{C}} \quad \ldots \quad (B)$$

For shafts having a single overhung crank, the form under the radical sign is to be multiplied by

$$s+\sqrt{s^2+1}.$$

For two cylinder single crank tandem engines the formula will therefore be:—

$$d=\sqrt[3]{\frac{P\,L\,(D_1^2+0.1\,D^2)(s+\sqrt{s^2+1})}{C}} \quad . \quad (C)$$

In those formulæ:

d = diameter of the after shaft bearing in inches.

n_1 = number of high pressure cylinders.

D_1 = diameter of each high pressure cylinder in inches. If there are several high pressure cylinders the diameters of which are not the same, $n_1\ D_1^2$ represents the sum of the squares of their respective diameters.

n = number of low pressure cylinders.

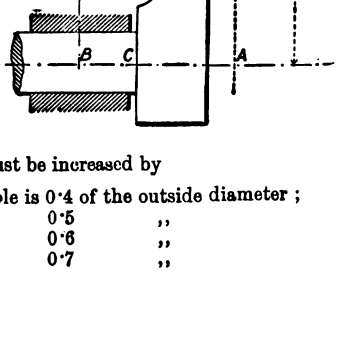
D = diameter of each low pressure cylinder in inches. If there are several low pressure cylinders the diameters of which are not the same $n\ D^2$ represents the sum of the squares of their respective diameters.

N.B.—For triple or quadruple expansion engines the intermediate cylinders do not come into account in the formulæ.

L = length of stroke in inches, common to all pistons.

P = boiler pressure above atmosphere in lbs. per sq. inch.

$s=\frac{a}{r}$ (see below). In order to determine a, B is supposed to be situated half-way the length of the bearing, unless the latter be longer than $1\frac{1}{2}$ times the diameter; in this case B C may be considered as being equal to $\frac{2}{4}$ of the diameter.



C = a constant, the values of which are given below for certain cases.

If the diameter of shaft be above 15″, it should be increased by an amount to be determined by the Administration; for built-up shafts, however, this latter increase will not be required.

For hollow shafts, the diameter must be increased by

1	per cent.	if the diameter of	the hole is	0·4 of the outside diameter;
2	,,	,,	,,	0·5 ,,
5	,,	,,	,,	0·6 ,,
10	,,	,,	,,	0·7 ,,

If the hole is under 0·4 of the outside diameter, no increase will be required.

The Administration may allow a reduction on the diameter in certain special cases, for instance in well-balanced engines with light moving parts or for very superior workmanship, etc. On the other hand, the Administration may require an augmentation for engines which differ much from the average proportions found in practice, thus, for instance, for engines having a comparatively small stroke; for compound engines the low pressure cylinder of which has a very large size compared with the high pressure cylinder, etc.

VALUES OF C IN CRANKSHAFT FORMULÆ (A), (B), AND (C).

Description of Engine.	Cranks. No.	Cranks. Angle.	No. of Cylinders.	No. of H. P. Cylinders.	No. of L. P. Cylinders.	Constant C.	Formula.
Simple or non-compound.	2	90° 100° 120° 140° 160–180°	2	...	2	4800 4370 3790 3450 3360	A
Two cylinder compound.	2	90° 100° 120° 140° 160–180°	2	1	1	3400 3090 2680 2440 2380	B
Triple expansion.	3	120°	3	1	1	3900	B
Four cylinder triple or quadruple.	4	90° Angles arranged to reduce maximum torsion moment.	4	1	2 or 1	4000 Not to exceed 4100	B

PROPELLER, TUNNEL AND THRUST SHAFTS.

The diameter of propeller shaft is to be found from the following formula:—

$$s = d + (1{\cdot}7\frac{D}{d} - 15) \times \frac{d}{100}.$$

Where δ = diameter of propeller shaft;
D = diameter of propeller; and
d = diameter of the crank shaft calculated from one of the formulæ (A), (B), or (C), all in inches.

It is recommended to fit the propeller shaft in such a way that it cannot move endways, if for some reason or other it has been uncoupled from the rest of the shafting.

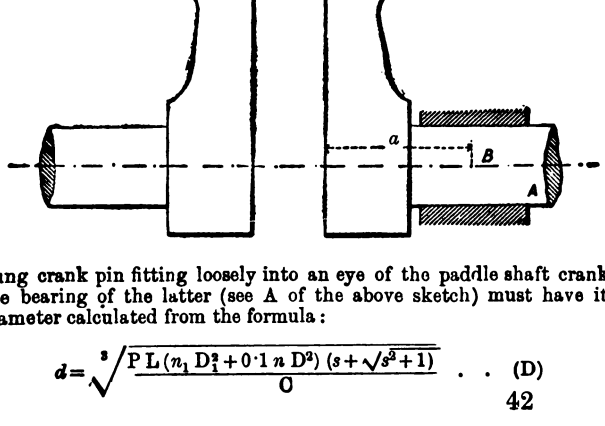
Liners fitted on propeller shafts to be tapered off at ends.

For tunnel shafts a reduction of 6 per cent. on the diameter of the crank shaft will be allowed.

The diameter of thrust shaft at the bottom of the collars, both between and immediately beyond these latter, to be equal to that of the crank shaft, and tapered off at each end to the smaller diameter of the body of the shaft. The thrust of the screw propeller must be taken up by an efficient thrust block, so as to prevent any fore-and-aft strain on the crank shaft.

Shafts for Paddle Steamers.

In side wheel steamers having double, triple, or quadruple expansion engines with an intermediate shaft, each end of which carries an over-



hung crank pin fitting loosely into an eye of the paddle shaft crank, the bearing of the latter (see A of the above sketch) must have its diameter calculated from the formula:

$$d = \sqrt[3]{\frac{P\,L\,(n_1\,D_1^2 + 0{\cdot}1\,n\,D^2)\,(s+\sqrt{s^2+1})}{0}} \quad . \quad . \quad (D)$$

42

where the letters have the same meaning as before, except that a, for determining $s=\frac{a}{r}$, is to be measured as shown in the above ketch, the point B being the middle of the bearing.

For two cylinder compound receiver engines with two cranks at 90°

C = 13,000 for navigation in smooth water.*
= 7,100 for coasting vessels.
= 5,700 for sea-going vessels.

For triple expansion engines with three cylinders and three cranks at 120°:—

C = 14,900 for navigation in smooth waters.*
= 8,150 for coasting vessels.
= 6,540 for sea-going vessels.

The diameter of the outer bearing of the paddle shaft and of the intermediate shaft to be submitted to the Administration or the Surveyors for approval. The same applies to other cases not dealt with in this paragraph.

Shafts for Turbine Engines.

In turbine engines, where I.H.P. is the estimated power transmitted by each shaft,

d = diameter of tunnel shafting in inches.
d = diameter of propeller shaft in inches.
D = diameter of propeller in inches.
R = number of revolutions per minute.

d and d_1 should be given by the following formulæ:

$$d=\sqrt[3]{\frac{70\times \text{S.H.P.}}{\text{R}}},$$

$$d_1=d+\frac{\text{D}}{160},$$

and rotor shaft is to have a diameter at the smallest part at least 5 per cent. greater than the tunnel shafting.

Steam Pipes.

Solid-drawn copper pipes to be of thickness $t=\frac{\text{PD}}{6400}+0{\cdot}05$.

Brazed copper pipes, $t=\frac{\text{PD}}{5440}+0{\cdot}05$.

Steel pipes to be of thickness, solid drawn, $t=\frac{\text{PD}}{7885}+0{\cdot}08$.

If welded, $t=\frac{\text{PD}}{6850}$.

If riveted, $t=\frac{a\text{PD}}{7885}+0{\cdot}08$, a being the strength of joint.

* Not now admitted.

P is boiler pressure in lbs. per square inch, and D is inside diameter in inches.

Spare Gear for Engines and Boilers.—For vessels employed in Mediterranean, in Continental coasting trades, and making long voyages, the following articles of spare gear are required to be on board :—

2 Top end bolts and nuts.
2 Bottom end bolts and nuts.
2 Main bearing bolts and nuts.
1 Set shaft coupling bolts and nuts (for one coupling).
½ Set feed pump valves.
½ Set bilge pump valves.
1 Set piston rings.
1 Set safety-valve springs.
5% of total number boiler tubes.
4% of total number condenser tubes.
½ Set of fire bars.
1 Canvas fire hose.
1 Set of spanners.
Plugs for boiler tubes.
12 Gauge glasses.
1 Steam gauge.

For vessels under French flag the following *additional* articles are required :—

1 Pair crank-pin brasses (or sufficient white metal and means for applying same).
½ Set bilge pump valves (metal).
1½ Set bilge pump valves (india-rubber).
½ Set bilge pump valve seats (if removable).
2 Sets gauge glasses.
½ Set fire bars.
1 Set electric hand lamps.
1 Safety lamp.
1 Set firing tools.

Also in vessels carrying coal or other dangerous cargo :—
4 Safety lamps.

And in vessels fitted with electric machinery, the necessary articles of spare gear according to the type of apparatus employed.

A further list is given in the Rules of articles that are *recommended* to be carried on board, and when these articles are so carried the vessel will be entered in the Register with the special mark ( ).

Qualities and Testing of Materials.—All material used in vessels for which the Special Survey mark is required must be tested, at the manufacturers' works if possible, and in the presence of a Surveyor to the Bureau Veritas, as hereinafter described, and all material that

Steel for Boilers.

Items.	Tensile and Elongation Tests.					Bending Tests.					
		Tensile Strength, Tons per sq. inch.		Elongation.		Cold Tests.			Quenching Tests.		
	Dimensions of Test Pieces.	Minimum.	Maximum.	For Tensile Strength of Tons.	Percentage in 8 ins.	Dimensions of Test Piece.	Internal Radius.	Angle.	Dimensions of Test Piece.	Internal Radius.	Angle.
Shell plates.	Working length 8 ins. Finished breadth 1½ ins.	26	32	26–32	24% to 20%	Strips 2 ins. wide.	Radius equal to thickness of plate.	180°	Strips 2 ins. wide.	Radius equal to 1½ times thickness of plate.	180°
				29	22%						
End plates.		25·5	30·5	30·5	20%			180°			180°
				25·5	25%						
Furnaces and Fireboxes.		24	29	29	21%			180°			180°
				25	25%						
				24	26%						

Steel for shell plates may be up to 35 tons tensile if the elongation is not less than 10 per cent.

Steel for Boilers (*continued*).

Long stays and Fire-box stays.	Working length 8 ins. or 4 ins. Diameter turned to $\frac{9}{10}$ in.	25·5	30·5	30	In 8 ins. 20%	In 4 ins. 25%	Short length of bar.	Radius equal to diameter of bar.	180°	Short length of bar.	Radius equal to 1½ times diameter of bar.	180°
				29	21%	25%						
				28	22%	26%						
				27	23%	27%						
				26	24%	27·5%						
				25	24·5%	28%						
Rivets.	Working length 8 ins.	24	29	29	25%		Short length of bar.	Doubled over.	180°	Short length of bar.	Doubled over.	180°
				28	26%							
				27	27%							
				26	27·5%							
				25	28%							
				24	29%							

Quenching tests to be made by heating the test pieces to a dull red and then quenching them in water at about 82° F.

A complete set of tests is to be made for each rolled plate and also for the long stays and for the fire-box stays of each boiler.

At least two sets of the following tests also to be made on the rivets of each boiler :—Cut short length equal to two diameters from rivet bar and compress cold to half length; cut similar piece, heat to dull red, and compress to quarter length, after which punch the pieces through with a punch having a diameter equal to ⅔ that of the rivet.

Steel Forgings and Castings, Cast-iron, Copper, Bronze, etc.

Items.	Tensile and Elongation Tests.					Cold Bending Tests.			Remarks.
	Dimensions of Test Pieces.	Tensile Strengths. Tons per sq. in.		Elongation.		Dimensions of Test Pieces.	Internal Radius.	Angle.	
		Minimum.	Maximum.	For Tensile of Tons.	Percentage.				
Steel forgings.	Working length 4 ins. diameter turned to $\frac{8}{10}$ in.	27	32	27	26%	1 in. square and 10 ins. long.	$\frac{3}{4}$ in.	180°	Two tests to be made for any number of forgings from the same charge. Test pieces to be cut cold from forging after annealing, and as far as possible from end of ingot at which metal was run in.
				28·5	25%				
				30	24%				
				32	23%				
Nickel steel forgings, 3·25% nickel, 0·35% carbon.	Do.	37	42	37–42	20%	Instead of bending test a piece 1 in. sq. placed on knife edges 6½ ins. apart must stand 15 blows from weight of 40 lbs. falling 9 ft.			Test pieces to be taken (one for each piece) after forgings have been annealed.
Steel castings.	Do.	Generally 28 38		28	16%	1 in. square	1½ in.	90°	Test piece 1 in. square placed on knife edges 6½ ins. apart must stand without fracture a series of shocks from a 40 lb. weight falling 3½ ft. to 5 ft. by 2 in. increments after each blow. One test from every charge and one for every large casting.
		Engine parts subject to alternating stresses 28 35		31	14%			90°	
				35	12%			90°	
		Crankshafts 28 30		38	10%			90°	

Steel Forgings, etc. (*continued*).

Cast iron.	Diameter turned to ⅞ in.	7½ (common) 11 (superior)	..	..	..	..	..		Pieces 1⅛ ins. square placed on knife edges 6½ ins. apart must stand without fracture a series of blows from a 27-lb. weight beginning at 10 ins. and advancing by 2 ins. to 16 ins. for common iron, and at 12 ins. and advancing by 2 ins. up to 18 ins. for superior iron.
Copper plates.	Working length 8 ins., finished breadth ⅞ in.	12·5	..	..	22%	strips 6 ins. long and 2 to 2½ ins. wide.	1½ times thickness of plate.	180°	Hot bending tests on 6-in. × 2-in. to 2½-in. strips must show that plates will bend double with faces touching without any sign of fracture.
Brass plates.		19	..	..	12%				
Copper pipes.	Working length 8 ins., finished breadth ⅞ in.	13	..	..	30%	..	..	..	End of pipe to be flanged over to ⅛ external dia. of pipe, but not exceeding 1½ ins. Radius of bend equal to thickness of tube: a piece of pipe 4 ins. long to be sawn through lengthwise and bent back reverse way to form a cylinder. Internal pressure test 430 lbs. per sq. in. Test pieces to be annealed.
Brass tubes.	..	.	..	..	..	.	..	..	Flanging test as for copper pipes (above). Diameter of tube to be increased ⅛ in. by drift. Piece of tube to be completely flattened. Hydraulic test 430 lbs. per sq. in. Test pieces to be annealed. Condenser tubes to be completely flattened. Hydraulic test 570 lbs. per sq. in. Test pieces to be annealed.
Turbine blades 72 Cu+ 28 Zn.	.	27·5	..	..	20	⅜ diam. ×4 ins. long.	..	..	..

satisfactorily withstands these tests is to be legibly stamped by the Surveyor. If any test of steel is unsatisfactory the Surveyor is entitled to condemn all material from that charge.

Steel must be made exclusively by the Siemens-Martin process; nevertheless steel produced by the electric furnace may be adopted with the sanction of the Administration.

Copies of lists of material giving all needful particulars are to be handed to the Surveyors.

In the United Kingdom, the tests of the British Standards Committee will be accepted if carried out in the presence of a Bureau Veritas Surveyor.

Boiler Tubes.—Material to have an ultimate strength lying between 22 and 25 tons per square inch, and an elongation of 22% on 8 inches if thicker than $\frac{1}{8}$ inch, and 20% if thinner. Test pieces to be $1\frac{1}{4}$ inches wide.

Material also to stand (cold) the following tests without cracking:—

1. End of tube to be expanded by a drift to 8% increase of outside diameter if over $\frac{5}{16}$ inch thick, and to 10% for thinner tubes.

2. A piece 4 inches long to be sawn through lengthwise at a reasonable distance from the weld, opened and rolled again in the opposite way to a cylinder.

3. A piece 4 inches long to be flattened, the weld being in the fold, until the distance between the sides is equal to the thickness for tubes of or over $\frac{5}{16}$ inch thick, and until the sides are close together for thinner tubes.

4. The end of the tube to be flanged at right angles into a rim of four times the thickness for tubes under $\frac{5}{16}$ inch thick and of $\frac{5}{8}$ inch for thicker tubes.

5. Hydraulic test to 560 lbs. per square inch, and each tube to be hammered whilst under pressure.

For brass castings the minimum strength is to be 11·5 tons per square inch, with an elongation of 8 per cent. on 4 inches.

For high tensile bronze the minimum strength is to be 28·5 tons per square inch, with an elongation of 18 per cent. on 4 inches.

In both cases the test pieces are to be turned to $\frac{5}{8}$ inch diameter.

Electric Lighting.

§ 1. *Dynamos* to be of an approved type; continuous currents are to be preferred. The engines should be fitted with a governor and situated in the engine-room or in a separate adjacent compartment and efficiently ventilated.

The dynamo circuits to be tested to 800 volts for 5 minutes.

§ 2. *Switch-Boards.*—The fittings on main switch-board should be mounted on slate or other incombustible material, placed near the dynamos, and be accessible from the back, if possible, unless all connections be established on the front of the switch-board. From this the main circuits should be led to auxiliary switch-boards for distributing the current to the various branches which may not be taken directly off a main cable.

A volt-meter for each installation and an ampère-meter for each dynamo to be supplied.

§ 3. *Leads* to be of copper wire having a conductivity of at least 98 per cent. Single wires should not be less than 20 S.W.G., with a sectional area of at least 1 square inch per 1250 ampères. The insulation on the leads should be absolutely watertight and be capable of bearing a temperature of 150° F. without being softened. All cables should have an insulation resistance of not less than 700 megohms per statute mile after twenty-four hours' immersion in sea water at a temperature of 60° F.

If alternating currents are used the insulation must be double that required for a continuous current of same voltage.

It is recommended to test the insulation of the electric plant before proceeding on a long voyage.

Leads should be accessible. In cabins they should be laid in battens with covers screwed on; where exposed to moisture they should be lead covered, and be armoured or protected by galvanized iron casings in cargo holds or wherever liable to be injured.

Watertight packings of an approved type must be fitted where leads pass through decks or watertight bulkheads, and the leads should be carefully protected from chafing against holes in beams, &c. All bends should be as easy as possible.

§ 4. *Joints* must be carefully made and insulated to the same degree as the cables, and should be in places always accessible.

Resin should be employed as a flux for soldering.

In the double-wire system joints in flow and return wires should not be opposite one another.

In single-wire plants the joints with hull must be accessible. Large cables should be secured in a copper plate bolted to the steel-work of the hull, which should be scraped bright at the contact, the area of contact being at least five times the sectional area of cables. A brass screw with non-oxidizable washer may be employed for single lamps and small cables.

§ 5. *Switches* should be made so as to act quickly, and be either full on or off without remaining in an intermediate position.

They should have large rubbing surfaces, and be so arranged that the friction takes off the oxide formed. A switch should be fitted to each main and branch circuit, and the more important ones arranged so as not to be tampered with by irresponsible persons. In places affected by moisture they should be fitted in watertight boxes having portable covers.

§ 6. *Fusible Cut-outs* should be fitted as a rule at the origin of each branch circuit, and be situated close to switch, those for the larger cables being placed on the switch-board. In double wire plants a cut-out should be fitted at the origin of each wire of each circuit. Like switches they should be placed in accessible positions and arranged so as not to be tampered with, be mounted on an incombustible base, and be fitted with a strong incombustible cover, which should be watertight where exposed to moisture. They must melt with a

current double the normal one, *i.e.* at 2500 ampères per square inch, and care must be taken to avoid mistakes in the size of fuses.

§ 7. *Lamps* to be of an approved pattern, strong, well insulated, and efficiently secured in their sockets.

Those in machinery space must be watertight and be protected by a glass globe with wire-netting.

In crew spaces, passages, holds, &c., they must be of a strong pattern efficiently protected by wire-guards. Deck lamps as well as side and masthead lamps must be perfectly watertight and be fitted with detachable connections.

§ 8. In Oil Vessels alternating current dynamos are strictly prohibited, and single-wire plants should not be fitted.

The insulating material must be such as not to be injured by oil or vapours.

The leads must not be run through the tanks, and no switches, cut-outs or joints may be placed in pump room.

All parts of circuits or fittings where sparks might be produced must be above the tank-deck, and situated where no gas can accumulate.

Arc lamps may not be used, and all incandescent deck and hold lamps are to be strongly protected by air-tight globes with wire-netting.

§ 9. *Compasses.*—The following precautions are recommended in order to lessen the deflections produced by electric-currents: compasses, especially the standard compass, should be at least 33 feet from any continuous current dynamo or electromotor, and 50 feet from alternating current machines. No single wire to be nearer a compass than 16 feet, and if this be not possible, the double wire or concentric system should be adopted in the vicinity of the compass. Chronometers also should be kept at a fair distance from dynamos and leads.

The influence of electric currents on compasses must be tested when these are being adjusted, with the vessel's head in any position and with all possible arrangements of current in the leads likely to affect the compass.

APPENDIX C.

Rules and Regulations in Force in U.S.A.

Being Extracts from those prescribed by the Board of Supervising Inspectors.

Steel Plates for Boilers shall be made by the Open Hearth process and not contain more than 0·04 per cent. of phosphorus, nor more than 0·04 of sulphur. When the tensile strength is less than 63,000 lbs. the minimum elongation shall be 25 per cent. for plates not exceeding $\frac{3}{4}$ inch, and 22 for those over. When the tensile is over 63,000 lbs. the elongation shall be 22 per cent. and 20 per cent.

Iron Plates not less than 45,000 lbs. tensile with 15 per cent. elongation.

Cylindrical Shells.—Factor of safety when single riveted must not be less than 6; when the longitudinal laps are double riveted working pressure may be 20 per cent. more—that is, the factor is 5.

Convex Drum Ends:—

$$\text{W.P.} = \frac{\text{T} \times \text{S}}{\text{R}}.$$

T is the thickness in inches.
S is one-fifth the tensile strength in lbs. per square inch.
R = one-half the radius to which this head is "bumped."

Concave Ends.—Working pressure is 80 per cent. of that for convex.

Flat Ends not exceeding 20 inches diameter:—

$$\text{W.P.} = \frac{\text{C} \times \text{T}^2}{\text{A}}.$$

T is the thickness in sixteenths of an inch.
A = one-half the area of end in square inches.
C = 112 for plates not exceeding $\frac{7}{16}$ inch; for those over, 120.

The radius inside at flange to be at least $1\frac{1}{2}$ ins. When holes are cut in boiler shells exceeding 6 inches diameter, doubling rings shall be fitted inside or outside to compensate fully. When working pressure does not exceed 75 lbs. the cast-iron strong flanges of mountings will be sufficient compensation.

Hydraulic Test Pressure to be 1·5 × working pressure.
Riveting as prescribed by the British Board of Trade.
Rounded Bottoms of Combustion Chambers :—

$$\text{W.P.} = \frac{50\,(300\text{T} - 2\text{L})}{\text{D}}.$$

T = thickness of plate in inches.
L = extreme length of plate forming bottom in inches.
D = twice the outside radius of curvature in inches.

Tube Plates, pressure allowed on when unsupported :—

$$\text{W.P.} = \frac{(\text{D} - d)\,\text{T} \times 27{,}000}{w \times \text{D}}.$$

D is the least horizontal pitch of tubes in inches.
d is the inside diameter of tubes in inches.
T the thickness of plate in inches.
w the extreme width of combustion chamber.

Furnaces.—Steel shall not exceed 67,000 nor be less than 54,000 lbs. ultimate tensile strength for corrugated furnaces, and not less than 58,000 lbs. for plain :—

$$\text{W.P.} = \frac{\text{C} \times \text{T}}{\text{D}}.$$

T the thickness in inches and not less than $\frac{5}{16}$ in.
D the mean diameter in inches.
C for Leeds Suspension 17,300 ; Morison 15,600 ; Fox 14,000 ; Purves 14,000 ; Holmes 10,000.

Adamson Furnace with flanges not less than 18 inches apart :—

$$\text{W.P.} = \frac{57{\cdot}6}{\text{D}}\{(18{\cdot}75 \times \text{T}) - (1{\cdot}03 \times \text{L})\}.$$

Spherical Top Furnaces :—

$$\text{Thickness in inches} = \frac{\text{P} \times \text{R}}{10{,}000} + 0{\cdot}12.$$

R is the radius of curvature in inches.

If made in more than one plate and c is the value of the joint as percentage of solid plate then

$$\text{Thickness will be } \frac{100 \times \text{T}}{c}.$$

Lap-welded Boiler Tubes up to 4 inches diameter shall be of mild steel (by any process) or charcoal iron.

A piece 2 inches long shall stand flattening to three thicknesses apart, have a flange turned over at right angles with a width of $\frac{1}{8}$ inch, both done cold.

Hydraulic pressure 1000 lbs. per square inch and then stand hammering. Tubes above 4 inches shall stand flattening to within three thicknesses, and a hydraulic pressure of 800 lbs. etc.

Seamless Tubes of Open Hearth steel, and when cold-drawn must be annealed. A piece 3 inches long to stand flattening to three thicknesses apart, and flanged as above. When hot finished no annealing, but same tests. Hydraulic test 1000 lbs. per square inch.

Their thickness must not be less than specified. Manufacturer to give a certificate that all these tests have been carried out.

Stays:—

$$\text{W.P.} = \frac{\text{A} \times \text{C}}{a}.$$

A the minimum cross section.
a the areas of surface supported.
C=9000 for tested steel over 1-inch diameter and annealed.
C=7500 ,, iron ,, ,, ,,

Bars for Stays.—Minimum tensile 58,000 lbs., elongation 28 per cent. in 2 inches; when more than 63,000 lbs., 26 per cent.

Girders on Combustion Chamber Tops and other flat surfaces:—

$$\text{W.P.} = \frac{\text{C} \times d^2 \times \text{T}}{(w-p) \times \text{D} \times \text{L}}.$$

w is the extreme width of box, etc., in inches.
p the pitch of supporting bolts in inches.
D the distance between girder centres in inches.
L the length of girder in feet.
d the depth of girder in inches.
C=550 when girder is fitted with 1 bolt; 825 when with 2 or 3 bolts; 917 when with 4 or 5 bolts; 963 when with 6 or 7 bolts; 990 when with more than 8.

Flat Surfaces.—Maximum stress allowable on:—

$$\text{W.P.} = \frac{\text{C} \times \text{T}^2}{p^2}, \text{ or } \text{C}\left(\frac{\text{T}}{p}\right)^2$$

T is thickness in sixteenths of an inch.
p the greatest pitch of stays in inches.
C=112 for screw stays with riveted heads, plates $\frac{7}{16}$ inch and under.
C=120 ,, ,, ,, ,, over $\frac{7}{16}$ inch.
C=120 when fitted with nuts and plates $\frac{7}{16}$ inch and under.
C=125 ,, ,, ,, over $\frac{7}{16}$ inch and under $\frac{9}{16}$.
C=135 ,, ,, ,, $\frac{9}{16}$ inch and over.
C=175 ,, ,, double nuts, without washers or doubling.
C=160 ,, ,, ,, and washers, etc., 0·5 × plate, and take T as 72 per cent. of combined thickness=T × 1·08.
C=200 when fitted with doubling plates 0·5 × T, covering the whole area and riveted to it inside or outside.

Fusible Plugs of bronze filled from end to end with pure (99·5 per cent.) tin with not more than 0·1 per cent. of lead nor more than 0·1 per cent. of zinc, $\frac{3}{4}$ inch external diameter, the tin being not less than $\frac{1}{2}$ inch. All tank boilers to have not less than two plugs. Vertical boiler plugs $\frac{3}{8}$ inch diameter with $\frac{1}{4}$ inch tin. Very elaborate rules for fixing plugs.

Water Gauges.—When these are on a stand pipe there must be an additional set of 3 cocks on the boiler shell. The lower gauge cock on boilers over 48 inches diameter must be 4 inches above chamber tops or flues. No need for these additional test cocks with water-tube boilers.

Double-ended boilers to have at least 3 test cocks and a water gauge at each end.

Steam Gauges.—One at least to each to indicate pressures up to 80 per cent. of the test pressure; that is 20 per cent. over the working pressure.

Safety Valves:—

$$\text{Area} = 0{\cdot}2074 \times \frac{W}{P} \times \text{sq. feet of grate.}$$

W = pounds of water evaporated per sq. feet of grate area per hour.
P = absolute pressure per sq. inch.

This means that area = weight of water evaporated per hour × 0·2074 ÷ absolute pressure.

Water-Tube Boilers.—Cylindrical drums having tube holes in their shells:—

$$\text{W. P.} = \frac{(p-d) \times T \times S}{p \times R}.$$

p is the pitch of the tubes in inches.
d is the diameter of holes in inches.
T the thickness of plate in inches.
S is $\frac{1}{6}$ of the tensile strength in lbs. of the plate.
R the radius of the shell in inches.
N the number of holes in a pitch.

N.B.—When the tubes are pitched diagonally, each diagonal ligament must be not less than $\frac{3}{5}$ of each longitudinal.

Hydraulic Test Pressure twice the working pressure.

Malleable cast iron or cast steel may be used for headers, boxes, bends, etc. Drum ends may be of cast steel, wrought iron, or steel.

Circumferential and horizontal seams of steam and water drums may be welded when the ends are spherical and they are properly annealed.

Seamless Copper or Brass Tubes not exceeding $\frac{3}{4}$ inch diameter may be used in water-tube boilers, and the drums may be also when the diameter does not exceed 14 inches and the thickness not less than $\frac{5}{8}$ inch; when drums are 12 inches diameter the minimum thickness may be $\frac{1}{2}$ inch.

Welding is allowed very freely both in construction and repairing of boilers.

Feed Water shall not be admitted to a marine boiler under 100° F. in temperature.

Main Steam Pipes.—Lap-welded wrought iron or steel, or solid-drawn steel, shall be as follows:—

$$\text{Thickness} = \frac{P \times D}{10{,}000} + 0{\cdot}125.$$

$$\text{W. P.} = \frac{(T - 0{\cdot}125)}{D} \times 10{,}000.$$

D is the inside diameter of pipe.
T the thickness in inches.

Steam and Water Pipes.—Up to 2 inches diameter shall be solid-drawn, those over 2 inches up to 30 inches shall be lap-welded. Threaded flanges are to be avoided, and on pipes over 5 inches diameter are not allowed.

Hydraulic tests. Up to 3½ inches diameter, 600 lbs.

Pipes 4 inches to 12 inches inside diameter and those 13 inches to 30 inches outside diameter: (*a*) steel pipes 2 inch length piece to be flattened to within three thicknesses, (*b*) iron pipes to have pieces in length equal to three thicknesses to be flattened to same extent. Tensile steel 50,000 lbs., with elongation 20 per cent. in 8 inches. Iron 40,000 to 48,000, elongation 12 per cent. in 8 inches, elastic limit 22,000 to 30,000.

Hydraulic test 4 inches to 30 inches diameter, 500 lbs.

Lap-welded or Solid-drawn Pipe of wrought iron or steel may be used for mud or steam drums not exceeding 15 inches diameter.

Copper Pipes.—No bend shall be allowed with a radius of curvature less than 1·5 × diameter:—

$$\text{Thickness} = \frac{P \times D}{6000} + 0{\cdot}0625 \text{ in inches.}$$

Cast Steel Pipes:—

$$\text{Thickness} = \frac{P \times D}{7000} + 0{\cdot}188.$$

Cast Iron, Semisteel, or Ferrosteel Pipes having an ultimate tensile not less than 20,000 may be used as follows:—

$$\text{Thickness in inches} = \frac{P \times D}{3000} + 0{\cdot}25.$$

Flanges of Pipes to be not less than four times the required thickness of pipe, plus ¼ inch. Hard brass, bronze, or other compositions of which 95 per cent. is copper, tin, and zinc, having an ultimate tensile of 30,000 lbs., may be used for all fittings up to 12 inches diameter, for all pressures up to 300 lbs., and temperatures to 425° F. Over these pressures and temperatures steel must be used.

APPENDIX D.

Lloyd's Rules relating to Use of Electric Light on Board Vessels.

Leads or Circuits.

1. The copper used in all wires or cables should have a conductivity of not less than that of the Engineering Standards Committee's standard, and the wires must be protected by tinning from the sulphur compounds present in the insulating material.

2. The sectional area of the copper wires in the cables should be in proportion to the current carried, and should not be less than as given in Table I., which is in accordance with the rules of the Institution of Electrical Engineers as revised in April, 1911.

3. Except for wiring fittings the sectional area of any copper conductor must not be less than that of No. 18 S.W.G. All copper conductors having a greater sectional area than No. 14 S.W.G. must be stranded.

4. The insulating material must be either vulcanized rubber of the best quality or it must be equally durable.

5. The insulation must be such that when the cables have been immersed in water for 24 hours it will, while still immersed, withstand 1000 volts for half an hour between the conductors and the water.

6. The insulation resistance should not be less than 600 megohms per statute mile at 60° F. after the cables have been immersed in water for 24 hours, the test being made after one minute's electrification at not less than 500 volts and while the cable is still immersed.

Joints.

7. Joints in branches, or of branches with leads of small circuits, must be made in properly constructed water-tight junction boxes, or should have the copper wires thoroughly soldered and the insulation carefully carried out, all the joints being made water-tight. Conductors of larger sectional area than $^7/_{18}$ S.W.G. must be soldered to proper lugs for connection. Joints in flow and return wires should not be made opposite one another. All joints should be in accessible positions, none being made in bunkers, cargo spaces, or spaces which may at any time be used for carrying cargo, stores, or baggage.

8. For soldering wires, the fluxes used must not contain acid or other corrosive substances.

9. Where practicable, the leads should be placed where they can always be accessible; if they are laid in wood battens the covers should be screwed on, not nailed, and care should be taken that the

casings are so arranged that water will not lodge in them. Cables which are properly covered with protective metal sheathing, or which are protected by galvanized wire armouring, may be unencased. They should, however, be secured by screwed clips, not by staples. All sharp bends in cables should be avoided.

Table I.

No. of wires and gauge in S.W.G. or in inches.	Nominal sectional area of conductors, sq. inches	Maximum current permissible. Ampères.	No. of wires and gauge in S.W.G. or in inches.	Nominal sectional area of conductors, sq. inches.	Maximum current permissible. Ampères.
Wires. 3/25	·0009	3·7	19/17	·046	70
Wires. 3/24	·0011	4·5	7/·097″	·050	74
Wires. 3/23	·0013	5·3	19/·058″	·050	74
1/18	·0018	7·2	19/16	·060	83
3/22	·0018	7·2	19/15	·075	97
7/25	·0022	8·6	19/14	·094	113
3/21	·0024	9·5	19/·083″	·100	118
1/17	·0025	9·8	37/16	·117	130
7/24	·0026	10·4	19/13	·125	134
3/20	·0030	12·0	37/15	·150	152
7/23	·0031	12·4	19/·101″	·150	152
1/16	·0032	12·9	37/14	·182	172
3/19	·0037	14·8	37/·083″	·200	184
1/15	·0041	16·3	37/13	·250	214
7/22	·0042	17·0	37/12	·300	240
1/14	·0050	19	37/·112″	·350	264
3/18	·0053	20	61/13	·400	288
7/21	·0055	21	61/·097″	·450	310
7/20	·0070	24	61/12	·500	332
7/19	·0086	28	61/·108″	·550	357
7/18	·0125	34	61/ 112″	·600	384
7/17	·017	40	61/·118″	·650	410
19/20	·019	43	91/·098″	·700	434
7/16	·022	46	91/·101″	·750	461
9/19	·023	47	91/·108″	·800	488
7/·068″	·025	50	91/·112″	·900	540
7/15	·028	53	91/·118″	1·000	595
9/18	·034	59	127/·101″	1·000	595
7/14	·035	60			

above sizes provide security against undesirable rise of tempera- For long leads larger wires will be required to prevent undue voltage.

10. All cables which are liable to be exposed to the weather or moisture should be lead-covered, or be otherwise specially protected. Where great heat is experienced, no wood casing should be used, but the cables should be protected by iron casings, or, if they are not exposed to mechanical injury, they may be armoured with galvanized wire and fastened to decks or bulkheads with screwed clips spaced not more than 12 inches apart.

11. If cables are led through cargo spaces, coal bunkers, or spaces which may at any time be used for carrying cargo, stores, or baggage, or which are not at all times accessible, they should be strongly protected against damage, preferably by iron casings. If they are led through metal tubes, these must be strongly secured, and should be fitted so that water cannot lodge in them.

Armoured cables may be used without casings or tubes provided they are strongly secured to the underside of decks or to bulkheads by screwed clips, and provided they are armoured in conformity with the standard of the Engineering Standards Committee, as shown in Table II.

Table II.

Diameter of cable to be armoured measured over lead covering.		Single wire armouring.	Double wire armouring.	Metal tape armouring.
Above.	Up to and including	Size of Galvanized Wire.	Size of Galvanized Wire in each layer.	Thickness of Metal Tape, two layers to be used.
inch.	inch.	inch. S.W.G.	inch. S.W.G.	inch.
...	·50	·072 No. 15	·056 No. 17	...
·50	·75	·092 No. 13	·072 No. 15	·03
·75	1·00	·104 No. 12	·080 No. 14	·04
1·00	1·25	·116 No. 11	·092 No. 13	·04
1·25	1·50	·128 No. 10	·104 No. 12	·04
1·50	1·75	·144 No. 9	·116 No. 11	·04
1·75	2·00	·160 No. 8	·128 No. 10	·04
2·00	...	...	...	·06

12. Where cables pass through beams, bulkheads, or other iron work, they should be led through special fittings of sheet lead, hard wood, or vulcanized fibre to prevent their being chafed, and where they pass through decks they should be led through metal tubes lined with wood or vulcanized fibre, and securely fastened to the decks, standing at such a height above the deck level that water cannot stand above them. Where cables pass through water-tight bulkheads the fittings must be made efficiently water-tight.

13. In vessels having spaces allotted alternately for passengers and

cargo, the lamp fittings in these spaces should be removable, and the terminals so arranged that they can be properly covered up with strong metal covers, or the whole of the fittings should be similarly provided with strong metal covers. The main switches and cut-outs should be outside these spaces, or if placed inside, they should be in strong iron boxes provided with iron covers, or otherwise securely arranged to prevent the fittings being tampered with.

Distribution.

14. The main switchboard should be fitted if possible in the dynamo room, to which all the main circuits throughout the ship should be brought, a switch and fuse being fitted thereon for each circuit. The auxiliary switchboards for further subdivision of the current should be placed in conveniently accessible positions, and each such switchboard should be similarly fitted with a separate switch and fuse for each sub-circuit. Fuses should be fitted to each lamp circuit when these are made with reduced size of wire. If vessels are wired on the double-wire system, fuses should be fitted to each cable of these circuits.

15. In cases where electric lights are used for the mast-head light and side lights, the switches controlling these lights should be placed in a position where they can be controlled by the officer of the watch, or other responsible person, and cannot be tampered with by other members of the crew, or by passengers, &c.

16. The switchboards should be of slate or other incombustible, non-conducting, and moisture-proof material. The switches should be on the quick-break principle, and should be so constructed that they must be either full "on" or completely "off," that is, they must not be able to remain in an intermediate position. They should have ample rubbing surfaces, and their conductivity should not be less than that of the wires connected to them, and they must be incapable of forming a permanent arc when breaking circuit.

17. Fuses should be fitted to each main or auxiliary circuit on the switchboards, as near as possible to the switches of these circuits. If the switchboard is not fitted near the dynamo, or if more than one dynamo may be used on any one circuit, then fuses should also be fitted to the main cable as near as possible to each of the dynamo terminals.

18. All fuses should be fitted in easily accessible places, and as near as possible to the commencement of the cables or wires they protect. They should be mounted on slate or other incombustible bases and be arranged so that the fused metal may not be a source of danger, and where they are fitted with covers these should be incombustible.

19. All fuses should be of easily fusible and non-oxidizable metal, and should be so proportioned as to melt with a current 100 per cent. in excess of that which the cables they protect are capable of carrying, as shown in Table I. The terminals must be spaced apart or screened, so that an arc cannot be maintained when the fuse is blown. Separate

single fuses and not double pole fuses must be used on circuits where the pressure exceeds 125 volts.

20. The fuses for each cable should be made of standard dimensions, so that a large fuse cannot be used for a small cable by mistake, or, if wire fuses are used, permanent instructions should be fitted on or near each switchboard giving particulars of the proper size of fuse for each circuit.

21. In shaft passages and in damp places, all lamp switches and fuses should be of a strong water-tight pattern, or should be placed in water-tight boxes having hinged or portable water-tight covers. No switches or cut-outs are to be placed in bunkers.

22. There should be no joints in the cables leading from the dynamo to the main switchboard, nor in those leading from the main to auxiliary switchboards, nor should branches to single lamps be taken off these cables.

23. A voltmeter should be supplied with each installation. If more than one dynamo is fitted, neither being capable of the whole of the output, an ampère meter should be supplied with each dynamo, unless each dynamo is protected by extra sensitive fuses.

Joints with Hull.

24. In vessels fitted on the single-wire system, all the joints with the hull should be placed in accessible positions. Those for single lamps or for small cables should be made with brass screws not less than three-eighths of an inch in diameter, carefully tapped into the iron or steel, having white brass washers between the wires and the vessel, or the wires should be soldered to brass-faced washers. For larger cables above $^{7}/_{18}$ S.W.G. and for the pole of dynamo the cable wires should be properly sweated into brass or copper shoes, which should be bolted to the vessel. The iron or steel where contact is made should be filed bright, and the area of contact should not be less than eight times the section of the copper of the cable.

In Vessels carrying Petroleum.

25. The single-wire system must not be adopted for any part of the installation. Switches and fuses must not be fitted in places liable to the accumulation of petroleum vapour or gas, and all lamps in places where it is possible for gas to accumulate must, with their holders, be enclosed in air-tight fittings of thick glass. All wires in such places are to be lead-covered, or the insulation of the cables employed is to be of such a nature as not to be affected by petroleum. No joints of cables, switches, or fuses should be fitted in the pump-room, but the wires for each lamp therein should be carried to the lamp from a distributing junction box placed outside the pump-room or companion.

The following paragraphs referring to the effect of the electric light installations upon the compasses are issued as suggestions, not as Rules.

Position of Dynamos and of Electric Motors.

26. The position and type of dynamos and electric motors should be such that the compasses will not be affected. Dynamos and large motors should be at least 30 feet from the standard compass.

Cables.

27. In vessels fitted with continuous-current dynamos, and wired on the single-wire system, no single cable should be carried within 15 feet of any compass, and cables conveying heavy currents should be fixed at still greater distance. If it is necessary to fix any cables within this distance, then for all parts of the vessel lighted from this cable the concentric or double-wire system should be adopted, the return-wire being carried as near the flow as possible, in the vicinity of the compasses.

Adjustment of Compasses.

28. The compasses should be adjusted with the dynamo not working, after which the vessel's head should be put upon the different courses, with the dynamo running at full speed, and on each course the indications of the compass should be noted with the dynamo running with open circuit and with all possible combinations of the current switched "on" and "off" all circuits passing near the compasses. These indications should be compared with those obtained with the dynamo stopped, and any serious deflections of the compasses remedied before the vessel sails.

APPENDIX E.

LLOYD'S REGISTER OF BRITISH AND FOREIGN SHIPPING.

Refrigerating Machinery and Appliances.

On the application of the owners of vessels fitted for carrying refrigerated cargoes, the Committee will authorize their Surveyors to survey the refrigerating machinery and appliances, and in those cases where the following conditions are complied with and a satisfactory report is received from the Surveyor, certificates of these Surveys will be issued, and the notation R.M.C. (*in red*) (*i.e.* Refrigerating Machinery Certificate) will be made against the vessel's name in the Society's Register Book, and in the special list of vessels fitted with refrigerating appliances. In cases in which the refrigerating machinery and appliances are constructed under the special survey of the Society's Surveyors and to their entire satisfaction, the notation ✠ R.M.C. (*in red*) will be made in the Register Book. The name of the maker and description and number of the refrigerating machines, whether single or duplex, and the refrigerating power of the machines will be recorded in the special list in the Register Book; as will also the number and capacity of insulated cargo chambers and the nature of the insulation and the method employed for cooling the holds.

1. The insulation must be sound and in good order and of efficient construction. The details of construction showing the amount and nature of the insulating material employed in the various parts are to be reported to the Committee.

Bilge suction and sounding pipes and ballast tank air and sounding pipes, passing through insulated spaces, should be well insulated to prevent their being frozen up. No sluice valves, scuppers or drain pipes are to be fitted which will permit drainage from spaces outside of the insulated chambers into the bilges of the insulated holds.

It is *recommended* that the woodwork of the insulation over tunnel tops be fastened with screws to facilitate the examination of this part, and that extra strong battens of American elm be fitted upon it under the hatches. Insulated removable portions are to be arranged in the bulkhead insulation, where required, to give easy access to sluice valves

and bilge suction roses. The bottoms, sides, and coamings of all insulated hatches and limbers should be painted to prevent decay.

Thermometer tube flanges and covers should be arranged so that water does not run down and freeze in them when taking the temperature.

Cargo battens should be provided for the floor or deck, and the sides of the chambers previous to loading the homeward cargo. Those for the sides of the chambers should be fastened, and should be at least 1½ inches in depth and 2 inches wide, one batten being placed over each frame or ground, the others being intermediately arranged. The battens for the floor and decks should be at least 2″ × 2″.

Where the brine system of refrigerating is employed, the brine circulating pipes and tanks should not be galvanized on the inside.

In cases where internally galvanized tanks and cooling pipes have been fitted, the brine cooling and return tanks, if closed, should be provided with two ventilating pipes communicating with the atmosphere. If the tanks are not closed, the cooling room should be efficiently ventilated.

2. The refrigerating machinery is to be of approved construction and of sufficient power to maintain the necessary low temperature in the cargo chambers in tropical climates when running eighteen hours per day. For cargo capacities of above 70,000 cubic feet the machinery is to be either duplex or in duplicate.

3. A sufficient amount of spare gear is to be supplied and stowed where it is readily accessible.

No spare gear will, however, be required in cases where two complete sets of refrigerating machines are fitted, *each* being of sufficient power to maintain the necessary low temperature in the cargo chambers in tropical climates when running eighteen hours per day, provided all the working parts of these machines are interchangeable.

When two similar machines are fitted, each connected to different cargo compartments, one set of spare gear suitable for either machine will suffice.

Where one single dry air machine is fitted to each compartment, the following will be required :—

1 crank shaft with eccentric sheaves, complete, or one half shaft if the halves are interchangeable.
1 piston rod and nuts for steam and air cylinders.
1 set of piston rod and connecting rod brasses.
1 piston, complete, for each steam and air cylinder.
1 cylinder cover for each pattern used in steam and air cylinders.
1 air pump bucket and rod.
1 circulating pump bucket and rod.
1 pair main bearing brasses, complete.
Main and cut-off valves for each steam cylinder.
Balance springs and rings for steam and air slide valves.
False valve face for each pattern fitted in steam cylinders, with screws.

1 eccentric rod for each pattern used.
1 eccentric strap for each pattern used.
1 slide valve spindle and nuts for steam and air cylinders, for each pattern used.
2 main bearing bolts.
1 set of connecting rod and piston rod bolts.
Full set of air valves and seats for air compressor.
1 set of inlet and outlet valves, and 1 set valve faces (if fitted) for air expansion cylinder with screws.
1 set of valves for air, circulating and feed pumps.
1 set of escape valve springs.
50 suction springs.
50 delivery springs.
50 buffer springs.
6 tubes and 24 ferrules for condenser.
6 tubes for cooler.
6 tubes for air drying chamber.
Assorted bolts, studs, and nuts.
1 set of lead-lined nuts for air expansion cylinder cover.
A quantity of packings and joint rings.

Where one duplex or two single dry air machines are fitted to each compartment, the following will be required:—

1 crank shaft with eccentric sheaves complete, or one half shaft if the halves are interchangeable.
1 piston rod and nuts for steam and air cylinders.
1 set of connecting rod and crosshead brasses.
1 piston for H.P. steam cylinder.
1 piston, complete, for air compressor; and 1 for air expansion cylinder.
1 set of piston springs for each steam cylinder.
1 cylinder cover for each pattern used in air compression and expansion cylinders.
1 air pump bucket and rod.
1 circulating pump bucket and rod.
Main and cut-off slide valves and spindles with nuts complete for H.P. steam cylinder.
Balance springs and rings for steam and air slide valves.
1 H.P. steam cylinder valve and valve face with screws.
1 eccentric sheave, strap, and rod for each pattern used.
1 slide valve spindle and nuts for steam and air cylinders for each pattern used.
2 main bearing bolts.
1 set of connecting rod and piston rod bolts.
Half set of air valves and seats for air compressor.
1 inlet and 1 outlet valve, and half set of valve faces (if fitted) for air expansion cylinder, with screws.
1 set of valves for air, circulating, and feed pumps.
1 set of escape valve springs.

20 suction springs.
40 delivery springs.
40 buffer springs.
6 tubes and 24 ferrules for condenser.
6 tubes for cooler and 6 for air drying chamber.
Assorted bolts, studs, and nuts.
Half set of lead-lined nuts for air expansion cylinder cover.
A quantity of packings and joint rings.

Where one single ammonia or carbonic anhydride compression machine is fitted :—

1 crank shaft with eccentric sheaves complete, or one half shaft if the halves are interchangeable.
Piston and rods complete with nuts for each steam cylinder and gas compressor.
1 air pump bucket and rod.
1 circulating pump bucket and rod.
1 pair main bearing brasses, complete.
1 set of connecting rod and crosshead brasses.
Main and cut-off valves for steam cylinders.
1 valve spindle for each pattern used and nuts complete.
1 eccentric strap and rod for each pattern used.
1 brine pump complete.
1 cover for each pattern used.
2 main bearing bolts.
1 set of connecting rod and piston rod bolts.
1 set compressor suction and delivery valves with springs and boxes, complete.
1 set of valves for air, circulating, feed, and brine pumps.
Crank shaft for fan engine.
1 steam piston and rod, etc., for fan engine, complete.
1 pair of connecting rod brasses for fan engines, with bolts, etc., complete.
1 set of blocks for making all leather packings used.
6 tubes and 24 ferrules for condenser.
Lengths and bends of piping of each size used, together with flanges, couplings, and screwing apparatus for effecting repairs.
1 gas regulating valve.
1 distributing and 1 collecting piece with multiple branches for coils for each pattern used. If these pieces are made of forged steel, no spare pieces are required.
Sundry valves, cocks, flanges, and fittings.
Assorted bolts, studs, and nuts.
Quantity of leather packings and joint rings.

For ammonia and carbonic anhydride compression machines, the lowing spare gear will be required, where one duplex or two single achines are fitted to each compartment :—

1 crank shaft, or one half shaft if the halves are interchangeable.
1 steam piston rod and nut for each pattern used.

1 piston for H.P. steam cylinder, with springs, complete.
1 set of piston rings for each steam cylinder.
1 set of piston rings for each size of compressor.
1 compressor piston rod and nuts, complete, for each pattern used.
1 air pump bucket and rod.
1 circulating pump bucket and rod.
Main and cut-off slide valves for H.P. steam cylinder.
Main and cut-off valve spindles and nuts for H.P. steam cylinder.
1 eccentric sheave, strap, and rod, for each pattern used.
1 brine pump complete.
1 cover for each end of gas compressor, except where screwed plugs are used.
2 main bearing bolts.
Half set of connecting rod and piston rod bolts.
Half set compressor suction and 1 delivery valve with springs and box, complete.
1 set of valves for air, circulating, feed, and brine pumps.
1 steam piston and rod, etc., for fan engine, complete.
1 pair of connecting rod brasses for fan engines, with bolts, etc., complete.
1 set of blocks and leather for making all leather packings used.
6 tubes and 24 ferrules for condenser.
Lengths and bends of piping of each size used, together with flanges, couplings, and screwing apparatus for effecting repairs.
1 gas regulating valve.
1 distributing and 1 collecting piece with multiple branches for coils for each pattern used. If these pieces are made of forged steel, no spare pieces are required.
Sundry valves, cocks, flanges, and fittings.
Assorted bolts, studs, and nuts.
A quantity of joint rings.

In cases where an independent circulating water pump is used, and its work *cannot* be performed by the main or auxiliary engines, a duplicate pump complete should be fitted.

In cases where an independent circulating water pump is used, and its work *can* be performed by the main or auxiliary engines, a pump bucket and rod should be carried, and half set of valves for water end.

In cases where an independent surface condenser with air, circulating, and feed pumps combined is fitted, and its work *cannot* be performed by the main engines:—

1 crank shaft with eccentric sheaves complete.
1 piston and rod complete for each pattern used.
1 eccentric strap and rod complete for each pattern used.
1 slide valve and spindle complete for each pattern used.
1 pump bucket and rod complete for each pattern used.
1 set of connecting rod and piston rod bolts and nuts.

1 set of valves for air, circulating, and feed pumps.
6 condenser tubes and 12 ferrules.

In cases where an independent surface condenser with air, circulating, and feed pumps combined is fitted, and its work *can* be performed by the main engines:—

1 pump bucket and rod complete for each pattern used.
1 set of connecting rod and piston rod bolts and nuts.
Half set of valves for air, circulating, and feed pumps.
6 condenser tubes and 12 ferrules.

Periodical Surveys.

4. In the cases of vessels engaged on voyages of more than three months' duration, a complete examination, as detailed in par. 9, is required every voyage. If this examination is made at other than the loading port a further examination is required at the loading port. (*See* par. 11.)

5. In the cases of vessels engaged on voyages of more than two and not more than three months' duration, a complete examination as above detailed is required at each alternate voyage, but at the intermediate voyage a modified examination as described in par. 10 will be sufficient, but the survey at loading port provided for in par. 11 should be held every voyage.

6. In vessels engaged on shorter over-sea voyages, the above examinations are to be held at least every three months, alternate examinations being as provided for in pars. 9 and 10, but the survey at loading port provided for in par. 11 should be held every voyage.

In the cases of vessels engaged on voyages of only a few days' duration, the complete examinations are to be held at least every three months, alternate examinations being as provided for in pars. 9 and 10, but the examination of insulation, etc., provided for in par. 11, instead of being held every voyage need only be held at intervals of four or six weeks, as may be approved in each special case.

7. *If in any case only part of the requisite examinations is held, the certificate will be endorsed with a statement of what is required to complete the survey.*

8. The date following the record R.M.C. *in red* indicates the date of the last examination of the Refrigerating Machinery and appliances as above mentioned.

When the periodical Surveys provided for in pars. 4, 5, and 6 are not held, the record R.M.C. will be expunged.

9. The complete periodical Survey required in par. 4 will consist of the following:—

The insulation throughout the holds is to be carefully examined and tested for dryness and fulness by sounding with a hammer and by boring. The test holes are to be afterwards efficiently closed. Special attention is to be paid to the spaces under the snow boxes, trunks,

and hatches where dampness may accumulate, to the sides under stringers and under decks and to the tunnel tops. All limber hatches are to be removed, the limbers cleared, and the suction pipes and roses, sluices and sounding pipes are to be examined. Hatches, air trunk-ways, and thermometer tubes with their connections and fastenings are to be examined, and where trunk-ways pass through watertight bulkheads, the watertight doors are to be examined and worked.

The trunk-ways should be as airtight as practicable and their fastenings should be secure.

The steam pipes, water pipes and connections, the crank shaft and bearings, connecting rods, steam and air cylinders, pistons, slides and valves, compressors and pistons, compressor rods and glands, surface condenser and air or gas coolers, circulating, air, feed and bilge pumps, are to be carefully examined and the condensers and coolers tested if deemed necessary.

The auxiliary machinery, where fitted, is also to be examined.

The spare gear is to be examined.

In dry air machines special attention is to be given to the condition of the air expansion cylinders, their pistons and valves. In other machines special attention is to be given to the condition of the compressors, including the pistons, rods, and glands, and to the expansion valve.

The refrigerator coils and their connections and the brine pipes and tanks, where fitted, are to be carefully examined and tested if deemed necessary.

Where the brine may escape to the bilges, the cement is to be examined.

The machinery is to be examined under working conditions, and tested on the snow box or refrigerators, the time and fall of temperature being noted.

It is *recommended* that the examination of the machinery under working conditions should be made upon the vessel's arrival at a home port, before the cargo is *fully discharged.* Where brine pipes are fitted they should be examined when under frosted conditions.

10. The examination required at alternate voyages in pars. 5 and 6 will consist of the following:—

The Insulation and Trunk-ways are to be surveyed in the same manner as is required for the complete examination detailed above. (*See* par. 9.)

Provided the machinery when tested under working conditions is found to be satisfactory, the following parts only will be required to be examined at this Survey, viz. :—

Steam valves, air pump, and circulating pump.
Crank shafts and bearings.
Air and other compressors and valves.
Expansion cylinder and valves in dry air machines.
Condenser water spaces.

Sea injection valves to be opened whenever the vessel is in dry dock.
Brine pipes in holds under frosted conditions.
The spare gear.

11. If the machinery and insulation have been surveyed and passed at a home port, the further survey required at a loading port will consist of an inspection to ascertain that the dunnage battens are in good order and that no damage has been sustained to the insulation prior to the loading of the refrigerated cargo, and also of a test of the refrigerating machinery under working conditions, the temperature in the hold being noted.

If the vessel loads at more than one port, one survey only at a loading port will be required, provided it includes the examination of all insulated spaces.

If there is no Surveyor to the Society available at the loading ports, or if there is not one obtainable from a port within a reasonable distance, this survey may be held at the port where the outward cargo is discharged. If there is no Surveyor to the Society at either of these ports, the Committee will accept the report of a survey held by a Surveyor appointed by Lloyd's Agent; or (in any case where there is no Lloyd's Agent) the report of a survey by a reliable Surveyor, if available; or (if no such Surveyor is available) a report, signed by two competent Engineers, of the vessel.

APPENDIX F.

STEERING GEAR.

I. Lloyd's Rules for Steering Gear.

Section 43. 1. Rudder Stops.—Suitable stops for the rudder should be securely fastened to the deck in way of the tiller or quadrant tiller. Where a suitable brake is fitted to the tiller or quadrant tiller, or where the steering quadrant is geared direct on to the steam steering engine, the deck stops may be dispensed with. The stops of the steam steering engines should be fitted at a smaller angle of helm than the rudder stops, so as to prevent excessive strains consequent on a rudder being forced against its stops.

2. **Spare Tiller.**—Vessels which have not two independent steering gears are to have spare tiller and gear ready for use when required.

3. **Independent Means of Steering.**—Where combined hand and steam steering gear is adopted, and in which both gears depend upon the efficiency of a keyed quadrant or tiller, independent means of steering must be provided.

Steamers above 250 feet in length are to be fitted with two independent steering gears, one of which must be a steam or other mechanical steering gear, and it is recommended that the two controlling wheels of the mechanical gear should be placed one at the gear and the other one on the navigating bridge.

4. **Protection of Steering Gear.**—In steamers above 250 feet in length, not having full poops or awning or shelter decks, the after steering wheel and gear are to be protected by a substantially constructed iron or steel deck-house or hood.

5. **Springs or buffers are to be fitted to all steam steering gears of steamers.**

6. **Steering Chains or Rods.**—The diameters of steering chains or rods are to be as given in Table XXV.* for the various diameters of rudder heads and the corresponding radii of quadrant tillers. Where the radius of quadrant or length of tiller adopted differs from that given in the Table, the diameter of steering chain is to be calculated from the following formula:—

$$d = \cdot 38\sqrt{\frac{D^3}{R}}$$

where d = diameter of chain in inches; D = diameter of rudder head in inches according to Table for rudder heads; R = radius of quadrant or length of tiller at the centre of the chain in inches.

* See "Lloyd's Rules and Regulations."

7. **Leads of Steering Chains.**—Care should be taken that the leads of the steering chains are made as direct as possible, sharp nips or bends being avoided.

8. The diameters at the centre of the chain of leading block sheaves are not to be less than sixteen times that of the steering chains, and the pins of the sheaves are not to be less than twice the diameter of the chains.

II. Board of Trade Requirements. Sec. 76.

A spare tiller (which has been properly fitted to the rudder head), relieving tackle, &c., should, in all foreign-going and home-trade steamships, be kept near the after steering gear ready for immediate service. In large steamships where the use of hand gear is impracticable and such gear is not provided, the spare tiller should be attached to the rudder head ready for immediate use, unless the working tiller is of special design and strength, in which case a spare tiller may not be required, but full particulars should be submitted for the Board's consideration. The steering gear, including chains, should be thoroughly overhauled at every survey, and taken to pieces and thoroughly examined at least once a year. The chains and blocks that are liable to interfere with or endanger the passengers or crew should be guarded by portable, but properly secured, guards.

With the view of relieving, as far as practicable, the rudders of vessels from severe and sudden shocks, springs have in some cases been fitted to the quadrant, or to the rods or chains at each side of the vessel, and the Board think that such fittings, or other efficient means, should be adopted, more particularly in the case of new vessels.

The Surveyors should note that the steam and exhaust pipes of steering engines in all new passenger steamships should be at least of the same internal diameter respectively as the steam and exhaust connections on the cylinders. The arrangement should be such that water will not readily lodge either in the cylinders or in the steam and exhaust pipes. Right-angled bends in the pipes should be avoided as much as possible, and the pipes should be used exclusively for the steering engines. When this is not the case, full particulars and sketches should be submitted to the Board for consideration.

Attention is also directed to a description of steam steering gear in which a part of the shaft, by which the helmsman actuates the controlling valve, passes through another shaft that is liable to be thrown out of line by the reaction of the spur gearing, and, consequently, is liable to jam the inside shaft to such an extent as to deprive the helmsman of the control of the steering gear. All steam steering gears should be carefully examined, and if any be found constructed in the manner described above, their use should be discouraged, and in any case they should not be approved, unless they have been tested from midship to hard over in both directions, and found satisfactory when the vessel is running at full speed.

APPENDIX G.

LLOYD'S REGISTER OF BRITISH AND FOREIGN SHIPPING.

Instructions to Surveyors regarding Tests, not Prescribed in the Society's Rules, for Steel and other Materials.

1. In cases where the material is not required to be in accordance with the Society's Rules, the Committee will approve of their Surveyors testing steel plates, sections and bars, iron and steel forgings and steel castings, to suitable specifications other than those stated in the Society's Rules, if these are mutually agreed upon by the manufacturers and purchasers. In the cases of rolled material tested under these conditions, the Brand must not be used, this being reserved exclusively for material conforming in all respects to the requirements of the Society's Rules.

2. The Committee will also approve of their Surveyors testing iron and steel tubes and pipes, copper and brass sheets, plates and tubes, and also similar material used in constructional work, as to which there are no tests mentioned in the Society's Rules, provided the tests actually specified are considered to be of a satisfactory character and are mutually agreed upon by manufacturers and purchasers.

In cases where no such tests have been mutually agreed upon, the following specifications for the different materials are suggested as being suitable for the mutual agreement referred to, viz. :—

Steel Forgings for Special Purposes.

In cases where it is desired to depart from the limits of strength of material provided for in the Rules of Lloyd's Register, viz. : from 28 to 32 tons per square inch, the Rules of the British Engineering Standards Committee are suggested, viz. :—

Tensile Strength.—A margin of 4 tons per square inch shall be allowed between the specific maximum and minimum tensile breaking strengths.

Elongation.—The sum of the percentage of elongation measured on a Standard test piece and of the actual tensile strength in tons per square inch shall in no case be less than 57.

Bend Tests.—Cold bend tests shall be made upon test pieces having

a rectangular section of 1 inch wide by $\frac{3}{4}$ inch thick. The test pieces shall be machined and the edges rounded to a radius of $\frac{1}{16}$ inch, and shall be bent over the thinner section. The tests may be made by pressure or by blows. The test pieces must withstand, without fracture, being bent through an angle of 180°, the internal radius of the bend being not greater than as specified below :—

Maximum Specified Tensile Strength of Forging.	Internal Radius of Test Piece after Bending.
	Inch.
32 tons per square inch	$\frac{1}{4}$
Above 32 and up to 36 tons per square inch . .	$\frac{3}{8}$
,, 36 ,, 40 ,, ,, . .	$\frac{5}{8}$

Lapwelded Iron or Steel Boiler Tubes.

Two per cent. of plain tubes and 2 per cent. of stay tubes are to be subjected to a hydraulic test of 750 lbs. per square inch.

Two per cent. of plain tubes are to be expanded cold at both ends by roller expanders to an increase of $\frac{1}{10}$th of their diameter. These tubes are then to be cut up, and two pieces, each 2 inches long, taken from each for crushing and bending tests.

A piece of each selected tube, 2 inches long, shall be placed on end, and must stand hammering down until it is reduced to $1\frac{3}{4}$ inches long without showing crack or flaw. A similar test piece from each selected tube is to be flattened till its narrowest width is two-thirds the original diameter.

In the case of stay tubes where the ends are made thicker than the body of the tube, the thickening is to be done by upsetting, not by any welding process.

All the tubes are to be inspected and gauged for both diameter and thickness.

Should any of the above tests fail, two further tubes shall be taken and the test repeated on a piece or pieces cut from each of them. If the repeat tests from both tubes are satisfactory, the batch of tubes represented shall be accepted ; but, if defects are again shown, the batch of tubes represented shall be rejected.

Charcoal Iron Lapwelded Boiler Tubes.

Two per cent. of plain tubes and 2 per cent. of stay tubes are to be subjected to a hydraulic test of 750 lbs. per square inch.

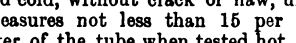
Two per cent. of each size of tube ordered shall be cut up to make the following tests :—

Each tube so selected must stand expanding at both ends, both

hot and cold, without crack or flaw, until the diameter of the bulged end measures not less than 15 per cent. greater than the original diameter of the tube when tested hot, and 10 per cent. greater when tested cold.

A piece of each selected tube, 2 inches long, shall be placed on end, and must stand hammering down until it is reduced to $1\frac{1}{2}$ inches long, without showing crack or flaw.

A similar piece from each selected tube must stand flattening until the interior surfaces of the tube meet without showing crack or flaw, thus:



In the case of stay tubes where the ends are made thicker than the body of the tube, the thickening is to be done by upsetting, not by any welding process.

All the tubes are to be inspected and gauged for both diameter and thickness. For the thickness, a tolerance of nothing negative and plus 10 per cent. may be permitted.

Should any of the above tests fail, two further tubes shall be taken and the test repeated on a piece or pieces cut from each of them. If the repeat tests from both tubes are satisfactory, the batch of tubes represented shall be accepted; but, if defects are again shown, the batch of tubes represented shall be rejected.

Seamless Steel Boiler Tubes.

Seamless steel boiler tubes may be finished hot or cold, as may be specified. They are to be well finished and clean inside and out, and free from laminations, surface defects, and rust, and unless otherwise specified are to be of uniform diameter throughout. Each tube is to be in a properly annealed condition when finished.

Any thickening required to the tube ends is to be done by upsetting, not by any welding process.

Two per cent. of the tubes are to be selected by the Surveyor for making the following tests:—

Expanding Test.—Both ends of the tubes are to withstand being expanded cold by a three-roller expander to the following increases of diameter:—

Thickness of Tube.	Increase of Diameter, per cent.
Under $\frac{1}{8}$ inch	$12\frac{1}{2}$
$\frac{1}{8}$ to $\frac{5}{16}$ inch	$9\frac{1}{2}$
Above $\frac{5}{16}$ inch	$6\frac{1}{2}$

After the expanding test, the selected tubes are to be cut up and the following tests are to be made from each:—

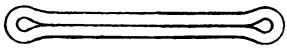
Tensile Tests.—Strips about $1\frac{1}{2}$ inches wide are to be cut from the tubes and flattened. These strips may be annealed after flattening. They are to have an ultimate tensile strength of not more than 26 tons per square inch in cold finished tubes, and 27 tons in those which are

hot finished, and an elongation of not less than 27 per cent. in a length of 2 inches in all cases.

Temper Test.—Strips about 1½ inches wide cut from the tubes and flattened, heated to a blood red, and cooled in water at about 80° F., are to stand being doubled over a radius of one-and-a-half times their thickness.

Crushing Test.—Pieces 2 inches long cut from the tubes are to withstand, when cold, being hammered down endwise until their length is reduced to 1 inch in the case of cold finished tubes, and to 1½ inches in the case of those finished hot.

Flattening Test.—Pieces 2 inches long, cut from cold finished tubes, are to withstand flattening by hammering, when cold, as follows, viz. :—

Under ⅛ inch in thickness . . .	Till the sides are close together thus : 
⅛ inch and up to $\frac{3}{16}$ inch . . .	Till the sides are brought to a distance apart equal to the thickness of the tube.
Above $\frac{3}{16}$ inch	Till the sides are brought to a distance apart equal to twice the thickness of the tube.

In the case of hot finished tubes of any thickness, the flattening shall be till the sides are brought to a distance apart equal to twice the thickness of the tubes.

Hydraulic Test.—The whole of the tubes are to be tested by hydraulic pressure to 1000 lbs. per square inch.

Should any of the above tests fail, two further tubes shall be taken, and the test repeated on a piece or pieces cut from each of them. If the repeat tests from both tubes are satisfactory, the batch of tubes represented shall be accepted; but, if defects are again shown, the batch of tubes represented shall be rejected.

The tubes are to be gauged for thickness and for diameter.

Tolerance as to Dimensions.—For thickness, a tolerance of nothing negative and of plus 10 per cent. for cold finished tubes may be permitted.

For hot finished tubes the tolerance may be nothing negative and plus 15 per cent. in tubes not over 10 feet in length, and of plus 20 per cent. in longer tubes.

For diameter, in all cases a tolerance of ½ per cent. greater diameter may be permitted. In cold finished tubes a tolerance of 1 per cent. smaller diameter, in tubes not more than 2 inches diameter, and of ½ per cent. in larger tubes, may be permitted.

In hot finished tubes the tolerance of less diameter may be 2 per cent. for tubes not more than 2 inches diameter, and 1½ per cent. in larger tubes.

Iron or Steel Steam Pipes.

These must be made of a thickness suitable for the diameter and the steam pressure at which they are to be used. If intended for use in vessels classed in Lloyd's Register, the thickness must be submitted to and approved by the Committee.

Lapwelded iron and lapwelded or seamless steel pipes are to be submitted to a hydraulic test of at least three times the intended working steam pressure.

The flanges, whether of iron or steel, are to be forged from the solid, and are to be screwed or riveted on, as may be specified. If the flanges are screwed on, the threads are to be vanishing.

In special cases the welding of flanges to the pipes may be approved, if particulars of the welding process are submitted and considered to be satisfactory.

All steam pipes should be properly annealed after the bending $\frac{\text{and}}{\text{or}}$ welding operations are completed.

Seamless and Brazed Copper Tubes.

The tubes must be clean, smooth, and free from surface defects. At least 2 per cent. of the tubes must be cut up for test.

Tubes may be ordered either annealed or hard drawn. All test pieces shall be annealed before testing.

The tubes must stand drifting, without showing either crack or flaw, until the diameter of the drifted end measures at least 25 per cent. greater than the original diameter of the tube.

A piece of the tube shall be flattened down until the interior surfaces of the tube meet, and then the flattened tube shall be doubled over itself close, that is, bent through an angle of 180°, the bend being at right angles to the length of the tube. This test should be made both hot and cold.

All tubes shall be subjected to an internal hydraulic test to such pressure as may be specified. If no pressure is specified a suitable pressure is given by—

$$\text{Pressure in lbs. per square inch} = 4500\frac{t}{D},$$

where t is the thickness and D the internal diameter in inches.

A tolerance of nothing negative and 5 per cent. positive may be allowed in the thickness.

For copper steam pipes, electro-deposited pipes must not be used. For large pipes, say 4 inches diameter and above, instead of the flattening test, the tests applicable to plates and sheets may be substituted upon strips cut longitudinally and transversely from the pipes.

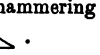
Copper Plates and Sheets.

The plates and sheets must be clean, smooth, and free from surface defects.

Each plate or sheet must be distinctly stamped with an identification mark.

The number of tensile and bend tests to be taken are as may be specified. For large firebox plates, a tensile and a bend test should be taken from each plate. The test pieces are to be annealed before testing.

The tensile test pieces, with a width of about 1½ inches and a length of 8 inches between gauge points, should show a tensile strength of not less than 14 tons per square inch, and an elongation of at least 35 per cent.

Bend tests, both hot and cold, should be capable of being flattened over on themselves both ways thus:  , without showing either crack or flaw on the outside of the bends. The cold bend test is afterwards to be drawn out cold by hammering to a feather edge, thus: drawn to .

Brass Condenser Plates, Condenser Tubes, &c.

Brass condenser plates are to be clean, smooth, and free from surface defects.

If tensile tests are required, the ultimate tensile strength should not be less than 22 tons per square inch, with an elongation of at least 10 per cent. measured on a Standard test piece.

Brass condenser tubes are to be seamless, of uniform diameter, and free from surface defects both inside and outside.

Each tube is to be tested by internal hydraulic pressure to 300 lbs. per square inch, the pressure being maintained while the outside of the tube is jarred along its length between suitable mallets.

The tubes are to be grouped in parcels of 100. Two of the tubes from each parcel are to be selected for the following tests:—

They are, without being previously annealed, to stand hammering on the end with a hand hammer, when held loosely, or jarring by being dropped when held horizontally from a height of 3 feet on a hard wooden floor. Pieces, 2 inches long, cut from either end of these tubes, are to be capable of being flattened till the diameter one way is 70 per cent. of the original. The tubes are also to stand heating to a dull red without fracture. They are afterwards to be cut open their whole length for an examination of the inner surfaces.

In the event of any of the above tests failing, two further tubes are to be selected from the same parcel and the failed tests are to be repeated. If these tests are satisfactory from both tubes, the parcel may be approved; but, if any failure again occurs, the parcel of tubes represented should be rejected.

High-Tension Brass Castings.

High-tension brass castings (including propeller blades) are to have a tensile strength of not less than 28 tons per square inch, with an elongation of not less than 15 per cent. measured on a Standard test piece.

High-Tension Brass Rolled Rods and Forgings.

High-tension brass rolled rods and forgings are to have a tensile strength of not less than 28 tons per square inch, with an elongation of not less than 28 per cent. measured on a Standard test piece.

Naval Brass Rods, Sheets, and Plates.

Naval brass bars are to be capable of (*a*) being hammered hot to a fine point, (*b*) being bent cold without annealing through an angle of 75° without fracture over a radius equal to the diameter of the bar.

Test pieces, at least 1½ inches wide, cut from Naval brass sheets and plates, must be capable of being bent cold through the angle specified over a radius equal to the thickness of the sample.

The tensile and bend tests for Naval brass are as follows:—

	Ultimate Tensile Tons per square inch.	Elongation on Standard Test Piece.	Bending Angle through
		Per cent.	°
Naval brass round and hexagon bars ¾ in. and under . .	26	20	75
Naval brass round and hexagon bars above ¾ in. . . .	22	20	75
Naval brass sheets ⅜ in. thick and under, annealed . .	26	30	180
Naval brass sheets ⅜ in. thick and under, not annealed .	26	25	135
Naval brass plates above ⅜ in. up to ½ in. thick . . .	26	20	135
Naval brass plates above ½ in. up to ¾ in. thick . . .	24	20	120
Naval brass plates above ¾ in. thick	22	22	90

APPENDIX H.

Lloyd's Register Rules for Diesel Engines.

Nominal Horse Power.

The following rule is to be used for determining the nominal horse power of Diesel engines in regulating the fees for their survey, viz. :—

$$\text{N.H.P.} = \frac{N \times D^2 \sqrt{S}}{80} \text{ in the case of single-acting engines of the 4-cycle type,}$$

$$= \frac{N \times D^2 \sqrt{S}}{40} \text{ in the case of single-acting engines of the 2-cycle type, and}$$

$$= \frac{N \times D^2 \sqrt{S}}{20} \text{ in the case of double-acting engines of the 2-cycle type,}$$

where D = diameter of cylinder in inches,
S = stroke of piston in inches in ordinary reciprocating engines,
= twice the stroke of piston in the case of engines of the "Junker" type,
N = number of cylinders.

Rules for the Construction and Survey of Diesel Engines and their Auxiliaries.

Section 1.—In vessels propelled by Diesel oil engines, the rules as regards machinery will be the same as those relating to steam engines, so far as regards the testing of material used in their construction and the fitting of sea connections, discharge pipes, shafting, stern tubes, and propellers.

Construction.

Section 2.—1. In vessels built under Special Survey and fitted with Diesel engines, the engines must also be constructed under Special Survey.

2. In cases of Diesel engines being built under Special Survey, the distinguishing mark ✠ will be noted in red, thus: ✠LMC or ✠NE.

3. In order to facilitate the inspection, the plans of the machinery

are to be examined by the Surveyors, and the dimensions of the shafts are to be submitted for approval.

4. The Surveyors are to examine the materials and workmanship from the commencement of the work until the final test of the machinery under full power working conditions; any defects are to be pointed out as early as possible.

5. Any novelty in the construction of the machinery is to be reported to the Committee and submitted for approval.

6. The auxiliary engines used for air compressing, working dynamos and ballast, or other, pumps, are also to be surveyed during construction.

7. In cases where the designed maximum pressure in the cylinders does not exceed 500 lbs. per square inch, the diameters of the crank shaft of the main engines are not to be less than those given by the following formula:—

$$\text{Diameter of crank shaft} = \sqrt[3]{D^2 \times (AS + BL)},$$

where D = diameter of cylinder,
S = length of stroke,
L = span of bearings adjacent to a crank, measured from inner edge to inner edge.

The values of (AS + BL) are as given in the following table:—

TABLE I.

4-Cycle Single-Acting Engine.	2-Cycle Single-Acting Engine.	Values of the Co-efficient.
4 or 6 cyls.	2 or 3 cyls.	·089 S + ·056 L
8 cyls.	4 cyls.	·099 S + ·054 L
10 or 12 cyls.	5 or 6 cyls.	·111 S + ·052 L
16 cyls.	8 cyls.	·131 S + ·050 L

For auxiliary engines of the Diesel type the diameters may be 5 per cent. less than given by the foregoing formula.

8. In solid forged shafts the breadth of the webs should not be less than 1·33 times and the thickness not less than 0·56 times the diameter of the shaft as found above, or, if these proportions are departed from, the webs must be of equivalent strength.

9. Where no fly-wheel is fitted, the diameter of the intermediate shaft must not be less than given by the formula:—

$$\text{Diameter of intermediate shaft} = \text{co-efficient} \times \sqrt[3]{D^2 \times S},$$

where D = diameter of cylinder,
S = stroke of piston,

and the value of the co-efficient is given by the following table:—

TABLE II.

4-Cycle Single-Acting Engine.	2-Cycle Single-Acting Engine.	Value of the Co-efficient.
4 cyls.	2 cyls.	·456
6, 8, 10, or 12 cyls.	3, 4, 5, or 6 cyls.	·436
16 cyls.	8 cyls.	·466

Where the stroke is not less than 1·2 times nor more than 1·6 times the diameter of the cylinder, (·735 D + ·273 S) may be taken instead of $\sqrt[3]{D^2 \times S}$.

10. In cases where fly-wheels are fitted, the following value of the co-efficient may be taken for determining the size of the intermediate shaft abaft the fly-wheel shaft.

TABLE III.

4-Cycle Single-Acting Engine.	2-Cycle Single-Acting Engine.	Value of the Co-efficient.
4 cyls.	2 cyls.	·405
6 ,,	3 ,,	·400
8 ,,	4 ,,	·409
10 ,,	5 ,,	·420
12 ,,	6 ,,	·427
16 ,,	8 ,,	·461

11. The diameter of the fly-wheel shaft must be at least equal to that of the crank shaft.

12. The diameter of the thrust shaft measured under the collars must be at least $\frac{11}{10}$ths that of the intermediate shaft. The diameter may be tapered off at each end to the same size as that of the intermediate shaft.

13. The diameter of the screw shaft must not be less than the diameter of the intermediate shaft (found as above) multiplied by $\left(\cdot 63 + \frac{\cdot 03\ P}{T}\right)$, but in no case must it be less than 1·07 T,

where P = the diameter of the propeller in inches,
T = the diameter of the intermediate shaft in inches.

The size of the screw shaft is intended to apply to shafts fitted with

continuous liners the whole length of the stern tube, as provided for in Section 18, paragraph 3, of the Rules for Engines and Boilers. If no liners are used, or if two separate liners are used, the diameter of the screw shaft should be $\frac{4}{10}$ths that given above.

The diameter of the screw shaft is to be tapered off at the forward end to the size of the thrust shaft.

14. If the designed maximum pressure in the cylinders exceeds 500 lbs. per sq. inch, the diameters of the shafting throughout must be increased in the proportion of $\sqrt[3]{\frac{\text{maximum pressure in lbs. per sq. inch}}{500}}$.

15. Where the cylinder liners are made of hard, close-grained cast iron of plain cylindrical form, accurately turned on the outside as well as bored on the inside so that their soundness can be ascertained by inspection, and their thickness at the upper part is not less than $\frac{1}{8}$th of the diameter of the cylinder, they need not be hydraulically tested by internal pressure. If, however, they are made of complicated form, the question of testing must be submitted.

16. The water jackets of the cylinders, and the water passages of the cylinder covers and pistons, must be tested by hydraulic pressure to 30 lbs. per square inch, and must be perfectly tight at that pressure.

17. The exhaust pipes and silencers must be water-cooled or lagged by non-conducting material, where risk of damage by heat is likely to occur.

18. The cylinders are to be fitted with safety valves loaded to not more than 40 per cent. above the designed maximum pressure in the cylinders and discharging where no damage can occur.

19. The air-compressors and their coolers are to be made so as to be easy of access for overhaul and adjustment.

20. In single-screw vessels, an auxiliary air-compressor is to be provided of sufficient power to enable the main engines to be kept continuously at work when the main compressor is out of action.

If the manœuvring gear is arranged so that the engines can be kept continuously at work with some of the cylinders out of action, the auxiliary compressor need only be of sufficient power to enable the engines to be kept at work under these conditions.

In twin-screw engines in which two sets of compressors are fitted, the auxiliary compressor must be of such size as to enable it to take the place of either of the main compressors. If in such engines each main compressor is sufficiently large to supply both engines, a smaller auxiliary compressor will be sufficient.

A small auxiliary compressor, worked by a steam engine, or by an oil engine not requiring compressed air, is to be fitted for first charging the air receivers.

21. At least one high-pressure air receiver is to be arranged with connections to enable it to be used for fuel injection, in case the working receiver of either main engine is out of use from any cause.

22. The circulating pump sea suction is to be provided with an efficient strainer which can be cleared inside the vessel.

Air Receivers.

Section 3.—1. Compressed-air receivers for starting air are to be supplied of sufficient capacity to permit of twelve consecutive startings of the engines without replenishment.

2. Cylindrical receivers for containing air under high pressure, used either for starting or for the injection of fuel in oil engines, may be made either of seamless steel or of welded, or riveted, steel plates.

3. **Quality of Material.**—If made of welded, or riveted, steel plates, the ordinary rules regarding steel material for boilers apply, which provide that where welding is employed, either in the longitudinal seams or at the ends, the material must have a tensile strength not exceeding 30 tons per square inch (Section 4, par. 7, Rules for Engines and Boilers). In these cases the welding must be lap welding; neither oxy-acetylene nor electric welding will be permitted.

4. In the case of seamless receivers, the rules for material will be the same as for boiler shells, but the permissible extension may be 2 per cent. less than that required with boiler plates.

5. **Tensile and Bend Tests** are to be made from the material of each receiver. When they are welded or riveted, the tests may be made, and the thicknesses verified, before the plates are bent into cylindrical form. In the cases of seamless receivers, the thicknesses must be verified by the Surveyor before the ends are closed in, and at this time the Surveyor shall select and mark the test pieces required from either of the open ends of the tube. The test pieces are to be annealed before test, so as to properly represent the finished material.

6. The permissible working pressure for welded or seamless receivers is to be determined by the following formulæ :—

Maximum working pressure in lbs. per square inch

$$= \frac{C \times S \times (T-2)}{D} \text{ for thicknesses of } \tfrac{1}{8} \text{ in. and above.}$$

$$= \frac{C \times S \times (T-1)}{D} \text{ for thicknesses below } \tfrac{1}{8} \text{ in.,}$$

where S = minimum tensile strength of the steel material used, in tons per square inch,
T = thickness of the material, in sixteenths of an inch,
D = internal diameter of cylinder, in inches,
C = co-efficient as per table :—

Co-efficient	77	for seamless receivers of	thickness of	$\frac{1}{8}$ in. and above,
,,	69	,, ,,	,,	,, below $\frac{1}{8}$ in.
,,	54	,, welded	,,	,, of $\frac{1}{8}$ in. and above,
,,	48	,, ,,	,,	,, below $\frac{1}{8}$ in.

7. For flat ends welded into the cylindrical shells, the thickness must not be less than

$$T = \frac{D}{17}\sqrt{P},$$

where T = thickness, in sixteenths of an inch,
D = internal diameter, in inches,
P = working pressure, in lbs. per square inch.

8. The permissible working pressure for receivers made of riveted steel plates is to be determined by the rules regulating the working pressure of boilers.

9. Each welded or seamless receiver shall be carefully annealed after manufacture, and before the hydraulic test.

10. Each welded or seamless receiver shall be subjected to a hydraulic test of twice the working pressure, which it shall withstand without permanent set.

11. Each receiver made of riveted steel plates is to be tested by hydraulic pressure to twice the working pressure for pressures up to 200 lbs. per square inch. Where higher working pressures are used, the test pressure need not be more than 200 lbs. per square inch above the working pressure.

12. All receivers above 6 inches internal diameter must be so made that the internal surfaces may be examined, and, wherever practicable, the openings for this purpose should be sufficiently large for access. Means must be provided for cleaning the inner surfaces by steam, or otherwise.

13. Each receiver which can be isolated must have a safety valve fitted, adjusted to the maximum working pressure. If, however, the air-compressor is fitted with a safety valve so arranged and adjusted that no greater pressure than that permitted can be admitted to the receivers, they need not be fitted with safety valves.

14. Each receiver must be fitted with a drain arrangement at its lowest part, permitting oil and condensed water to be blown out.

Pumping Arrangements.

Section 4.—1. The requirements of the pumping arrangements for the various holds, double bottoms or other ballast tanks, &c., are to be the same as required in steam vessels of the same size.

2. The engines are to be fitted with two bilge pumps, which are to be so arranged that either can be overhauled while the other is at work. In twin-screw vessels one bilge pump upon each engine will be approved. These pumps are to be arranged to draw from all compartments. Independent power-driven pumps may be fitted in lieu of these, if desired.

3. A steam pump, or equivalent power-driven pump, is also to be provided with connections to enable it to draw from all compartments

and from the sea. It must be arranged to discharge overboard and also on deck to the fire service pipes. It must have at least one suction to the engine-room bilge distinct from those connected with the bilge pumps, so that it may be used for pumping from the engine room when the bilge pumps are being used upon other parts of the vessel.

4. In addition to the above, where water ballast is used, the water-ballast pump must have one direct suction from the engine-room bilges. (This is in lieu of the bilge injection required with steam engines.)

General.

Section 5.—1. For the ordinary fuel tanks the requirements of Section 49 will apply. The daily service and other separate tanks must be tested, with all their fittings, with a head of water 12 feet above their highest points. They must be fitted with air pipes discharging above the upper deck. If they are fitted with glass gauges for indicating the quantity of oil contained in them, arrangements must be made for readily shutting off the gauges in the event of the breakage of the glass, and from preventing any damage from leakage of oil.

2. Special attention must be given to the ventilation of the engine room.

3. If the auxiliaries are worked by electricity, the cables in connection with them must be in accordance with the rules for cables for electric light.

4. It is recommended that all pipes conveying fuel oil should, as far as possible, be made of steel or iron, rather than copper, owing to the rapid corrosion of copper pipes when using oil containing sulphur.

Spare Gear.

Section 6.—The articles mentioned in the following list will be required to be carried, viz. :—

1 cylinder cover complete for the main engines, with all valves, valve seats, springs, &c., fitted to it.

In addition, one complete set of valves, valve seats, springs, &c., for one cylinder of the main and of the auxiliary Diesel engines, and fuel needle valves for half the number of the cylinders of each engine.

1 piston complete, with all piston rings, studs, and nuts for the main engines.

In addition, one set of piston rings for one piston of the main and of the auxiliary Deisel engines.

1 complete set of main skew wheels for one main engine.

2 connecting rod, or piston rod top-end bolts and nuts, both for the main and the auxiliary Diesel engines.

2 connecting rod bottom-end bolts and nuts, both for the main and for the auxiliary Diesel engines.
2 main bearing bolts and nuts, both for the main and for the auxiliary Diesel engines.
1 set of coupling bolts for the crank shaft.
1 set of coupling bolts for the intermediate shaft.
1 complete set of piston rings for each piston of the main and of the auxiliary compressors.
1 half set of valves for the main and for the auxiliary compressors.
1 fuel pump complete for the main engine, or a complete set of all the working parts.
1 fuel pump for the auxiliary Diesel engine, or a complete set of all working parts.
1 set of valves for the daily fuel supply pump.
1 set of valves for the water circulating pumps.
1 set of valves for one bilge pump.
1 set of valves for the scavenge pump, where lift valves are used.
A quantity of assorted bolts and nuts, including one set of cylinder cover studs and nuts.
Lengths of pipes suitable for the fuel delivery and the blast pipes to the cylinders, and the air delivery from the compressors to the receivers, with unions and flanges suitable for each.

Periodical Surveys.

Section 7.—1. The engines are to be submitted to survey annually, and, in addition, are to be submitted to a Special Survey upon the occasion of the vessels undergoing the Special Periodical Surveys Nos. 1, 2, and 3 prescribed in the Rules, unless the machinery has been specially surveyed within a period of twelve months, in which case the Annual Survey will be sufficient. The boilers, if fitted, are to be subjected to the same surveys as required by Section 19 of the Rules for Engines and Boilers.

2. **Special Surveys.**—At these special surveys, the main engines and the auxiliary engines are to be examined throughout, viz.:—All the cylinders, pistons, valves and valve gears, connecting rods and guides, pumps, crank, intermediate, and thrust shafts, propellers, stern bushes, sea connections and their fastenings, are to be examined. The air-compressors are also to be examined. The air-receivers are to be cleaned and examined and, if necessary, tested, as provided for in paragraph 3 of this section.

3. **Annual Surveys.**—The whole of the parts of the engines which the engineers of the vessel open up for adjustment and overhaul should be examined and reported upon. The Survey must include, for each main engine, the examination of at least 2 pistons, 2 cylinder covers and their valves, 2 connecting rods and their brasses, both top and bottom ends, 2 of the main bearings and crank shaft journals, and

1 of the tunnel bearings. If these are all satisfactory, their condition may be taken as representing that of the other similar parts.

In the auxiliary Diesel engines, a similar course must be adopted, but in this case one of each of the parts mentioned of each engine will be sufficient, if found to be satisfactory.

The valve gears of all the Diesel engines should be examined, as far as practicable, without complete dismantling.

The air receivers must be examined internally if possible, and, together with the air pipes from the compressors, must be cleaned internally by means of steam, or otherwise. If the air-receivers cannot be examined internally, they must be tested by hydraulic pressure to twice the working pressure at each alternate Annual Survey, attention being specially given to the welding of the ends and of the longitudinal joints.

Some Ships fitted with Oil Engines—Diesel System Type.

Name.	Date.	Gross R. Tonnage.	No. of Screws.	Engines.		Cylinder.		
				N.H.P.	Cycle.	No.	Diam.	Stroke.
							ins.	ins.
Annam . .	1913	5296	two	606	four	16	23·3	31·5
Arabis . .	1914	3697	two	302	two	8	16·5	33·9
Ares . . .	1914	3783	two	375	four	12	20·5	35·5
A. von Gwinner .	1912	3051	two	385	two	8	17·3	20·5
Artemis . .	1914	3803	two	385	four	12	20·5	35·5
Bandon . .	1909	3409	one	324	four	6	26·4	39·4
Bayard . .	1915	2900	two	288	four	12	19·4	26·0
Malaya . .	1921	9050	two	854	four	6	29·0	45·3
Calgary . .	1912	1639	two	53	two	8	11·4	16·9
Chile . .	1915	5570	two	568	four	12	24·8	37·8
Fiona . .	1914	5219	two	835	four	12	29·9	43·5
Glengyle . .	1915	6225	two	534	four	12	24·8	33·5
Glenpool . .	1913	5459	two	586	two	12	18·7	31·5
Hamlet . .	1915	5093	two	497	two	6	23·6	35·4
Juno. . .	1912	2345	two	228	four	6	22·0	39·4
Jutlandia . .	1912	4874	two	468	four	16	20·9	28·8
Kangaroo . .	1915	4348	two	398	four	12	22·0	29·8
Pangan . .	1909	3409	one	324	four	6	26·4	39·4
Rolandseck .	1912	1663	one	361	two	6	20·0	36·3
W. E. Riedemann	1914	9800	two	906	two	12	22·6	39·4
Wotan . .	1913	5703	one	550	two	6	23·4	43·3

The pumps and air-compressors must be examined and tried under working conditions. If found to be satisfactory, they need not be dismantled.

The manœuvring of the engines must be tested under working conditions.

If the examination reveals any defects, the Surveyor should recommend such further opening up as he may consider to be necessary.

4. **Record of Survey.**—If the various parts mentioned in paragraphs 2 and 3 are all found to be in a satisfactory condition and the Surveyor finds that the machinery generally is in good order, he should recommend the vessel to have a fresh record of LMC.

5. **Survey of Screw Shafts.**—Screw shafts are to be drawn and examined at intervals of not more than two years.*

* On the application of Owners, the Committee will be prepared to give consideration to the circumstances of any special case.

APPENDIX I.

Lloyd's Rules for Screw Shafts and Stern Tubes.

Section 8.—1. All shafts are to be turned all over and are to be examined when rough turned and when finished. In the case of screw shafts scrap steel is not to be used, and in no case is a mixture of scrap iron and scrap steel to be employed. It is recommended that screw shafts be made of ingot steel or forged from blooms made from rolled iron bar of good fibrous quality.

2. Gauges of an approved description for testing the truth of the crank shafts are to be supplied with all new engines, and adjusted in the presence of the Surveyor.

3. The length of the stern bush is to be at least four diameters of the shaft. It is recommended that the shaft liner should be continuous the whole length of the stern tube, and that the after end should be tapered in thickness and made watertight in the propeller boss. If the liner is made in two pieces the joint should be burned. If the liner does not fit tightly at the part between the bearings in the stern tube, the space between the shaft and the liner should be charged or "forced" with a plastic material insoluble in water and non-corrosive. If two liners are used, it is recommended that they be tapered in thickness at the ends, and that the shaft should be lapped or protected between the liners. In this case, and also if no liners are used, the diameter of the shaft should be $\frac{23}{20}$ths of that required for a shaft with a continuous liner.

4. For dimensions of shafts, see the formula.

APPENDIX K.

The distances to various Ports from Gibraltar and Sunderland in Nautical Miles.

Ports.	Distance.	Ports.	Distance.	Ports.	Distance.
From Gibraltar *to*—		Alexandria	1805	Hull	123
		Azores	950	Leith	125
Algiers	410	Boston, U.S.A.	3080	Limerick	827
Bermuda	2965	Buenos Ayres	5410	London	307
Barcelona	530	Cape Town	5130	Lynn	145
Cadiz	64	Charleston	3705	Middlesbro'	27
Carthagena	240	Cape Verde Is.	1550	Newport(Mon.)	730
Cette	670	Cape Horn	6500	Plymouth	513
Constantinople	1780	Finisterre	556	Sharpness	750
Cuxhaven	1550	Havana	4230	Southampton	393
Dover	1239	Huelva	130	Swansea	700
Finisterre	556	Jamaica	3960	Yarmouth	173
Genoa	865	Lisbon	300		
Halifax	2670	Las Palmas	850	Christiania	536
Holyhead	1221	Madeira	600	Copenhagen	607
Leghorn	870	New York	3250	Cronstadt	1277
Liverpool	1290	New Orleans	4815	Dantzig	883
London	1326	Newport News	3415	Helsingfors	1145
Mahon	622	Oporto	460	Kiel	695
Malaga	70	Quebec	3400	Memel	898
Malta	980	Rio Janeiro	4240	Malmö	615
Marseilles	690	Trinidad	3430	Revel	1113
Messina	995	Panama	4340	Riga	1062
Naples	970			Skaw	464
Oran	222	*From* Sunderland *to*—		Stockholm	995
Palermo	904				
Port Said	1930	Aberdeen	150	Archangel	1755
Rome	960	Barry	712	Amsterdam	280
Syracuse	1020	Belfast	605	Antwerp	320
Tangiers	30	Bristol	742	Bergen	437
Tarragona	490	Cardiff	720	Boulogne	305
Tunis	780	Cork	724	Bremerhaven	366
Ushant	940	Dover	267	Calais	276
Valencia	390	Dublin	792	Dunkirk	285
		Dundee	125	Ghent	305
		Falmouth	542	Gothenburg	505
Almeria	150	Glasgow (N.)	645	Hamburg	432
Alicante	310	Holyhead (N.)	707	Rotterdam	295

N.B.—(S.C.) means *vid* Suez Canal.

The distances of various Ports from Cardiff in Nautical Miles.

Ports.	Distance.	Ports.	Distance.	Ports.	Distance.
Belfast	310	Smyrna	2,776	Colon	4632
Cork	215	Suez	3,161	Maranham	3898
Glasgow	400	Trieste	2,806	Monte Video	6139
Hartlepool	705	Venice	2,800	Para	4058
Hull	638			Pernambuco	3940
Liverpool	270	Batoum	3,515	Rio Janeiro	5027
London	526	Constantinople	2,930	Rio Grande do Sul	5751
Southampton	338	Galatz	3,274		
Tynemouth	728	Odessa	3,272	Santos	5227
		Taganrog	3,519		
Copenhagen	1108			Barbadoes	3580
Cronstadt	1776	Aden	4,489	Belize	4510
Kiel	1185	Akyab	7,651	Cartagena	4310
Stettin	1267	Bombay	6,154	Demerara	3857
Stockholm	1498	Bushire	6,281	Greytown	4657
Uleaborg	1910	Calcutta	7,612	Havana	4025
		Colombo	6,606	Jamaica	4034
Archangel	2221	Hong Kong	9,716	Martinique	3555
Gothenburg	1000	Karachi	5,979	New Orleans	4559
Hamburg	821	Madras	6,872	St Vincent	3615
		Nagasaki	10,666	Tampico	4846
Bordeaux	542	Penang	7,879	Trinidad	3780
Bilbao	560	Rangoon	7,753	Vera Cruz	4840
Cadiz	1093	Shanghai	10,466		
Lisbon	882	Singapore	8,186	Baltimore	3285
Oporto	710	Yokohama	11,094	Belle Isle	1962
Vigo	665			Boston	2780
		Adelaide (S.C.)	10,710	Cape Race	1972
Algiers	1562	Antofagasta	9,397	Charleston	3437
Barcelona	1664	Auckland	11,768	Halifax	2440
Genoa	2020	Brisbane (S.C.)	11,961	New York	3000
Gibraltar	1153	Hobartown (S.C.)	11,195	Prince Edward Island	2433
Marseilles	1844				
Palermo	2057	Mauritius (S.C.)	9,571	Portland	2745
Tunis	1943			Quebec	2812
		Melbourne (S.C.)	11,035		
Alexandria	2943			Azores	1278
Athens	2620	Sydney (S.C.)	11,521	Cape Town	5998
Beyrout	3192			Cape Verde	2345
Brindisi	2427	Bahia	4,370	Delagoa Bay	7148
Fiume	2778	Bermuda	2,850	Lagos	4146
Malta	2133	Buenos Ayres	6,249	Madeira	1297
Port Said	3072	Cape Horn	7,222	Sierra Leone	2885

The distances of various Ports from Glasgow in Nautical Miles.

Ports.	Distance.	Ports.	Distance.	Ports.	Distance.
From Glasgow *to*—		Gothenburg	884	Havana and Jamaica	4200
		Hamburg (N.)	951	Vera Cruz	5015
Aberdeen	520	Bilbao	805		
Barry Dock	392	Bordeaux	787	Boston, U.S.A.	2738
Belfast	115	Cadiz	1,340	Charleston (S.C.)	3417
Bristol	422	Corunna	810	Halifax, N.S.	2388
Cardiff	400	Lisbon	1,118	Montreal, Canada	2907
Cork	350				
Deal	696	Barcelona	1,908	New York	3120
Dundee	580	Gibraltar	1,401	Portland (Me.)	2700
Falmouth	430	Marseilles	2,085	Pictou	2476
Greenock	18			Quebec	2735
Hull (N.)	745			Sable Island	2245
Limerick	460	Alexandria	3,185	St John's (N.F.)	1907
Liverpool	220	Dardenelles	3,028	St John (N.B.)	2591
London	775	Malta	2,375		
Plymouth	462	Port Said	3,325		
Southampton	582			Azores (S. Michael)	1520
		Bombay	6,396	Bonny	4498
				Bathurst	2728
Copenhagen	984			Canaries (Las Palmas)	1775
Cronstadt	1652	Hong Kong	9,958		
Dantzig	1250			Cape Town	6240
Kiel	1061			Delagoa Bay	7390
Stockholm	1380	Callao	10,382	Lagos	4388
		Sydney	11,763	Madeira	1540
		Valparaiso	9,111	Teneriffe	1765
Archangel	1981			Zanzibar (*vid* S.C.)	6436
Bremerhaven (N.)	901	Buenos Ayres	6,491		
		Rio Janeiro	5,280		

The distances of various Ports from Hull in Nautical Miles.

Ports.	Distance.	Ports.	Distance.	Ports.	Distance.
Aberdeen	275	Elsinore	610	Iceland	1275
Barrow	725	Gefle	1148	Hammerfest	1335
Belfast	805	Helsingfors	1168	Havre	300
Barry	630	Heårnosand	1248	Ostend	194
Bristol	660	Kiel	718	Rotterdam	212
Cardiff	638	Königsberg	930	St Malo	400
Cork	660	Memel	920	Bilbao	840
Dover	190	Malmö	638	Bordeaux	825
Dublin	710	Norkoping	966	Brest	540
Dundee	240	Pillau	902	Corunna	860
Falmouth	450	Revel	1135	Cadiz	1398
Finisterre	890	Riga	1085	Huelva	1375
Glasgow (N.)	745	Rostock	735	Lisbon	1156
Hartlepool	110	Skaw	490	Oporto	1012
Harlingen	225	Stettin	793	Santander	825
Leith	240	Stockholm	1019	St Nazaire	679
Limerick	816	Stralsund	718	Vigo	970
Liverpool	760	Sundswall	1199	Villa Real	1345
Lizard	455	Swinemünde	753	Algiers	1865
London	225	Travemünde	706	Alicante	1755
Lynn	70	Uleaborg	1431	Almazaron	1681
Middlesbro'	110	Wiborg	1265	Almeria	1599
Milford Haven	588			Barcelona	1966
Newport(Mon.)	648	Archangel	1870	Bona	2101
Plymouth	420	Amsterdam	209	Carthagena	1693
Southampton	325	Antwerp	230	Cette	2118
Sunderland	123	Arendal	460	Genoa	2305
Start Point	392	Bergen	500	Gibraltar	1440
St Catherine's (I. Wight)	300	Bremerhaven	340	Girgenti	2363
		Calais	200	Leghorn	2310
Swansea	618	Christiania	582	Malaga	1515
Troon (N.)	750	Christiansand	423	Marseilles	2146
Tyne	130	Christiansund	720	Messina	2471
Ushant	500	Dunkirk	200	Naples	2433
Waterford	626	Fredrikshald	548	Oran	1685
Whitby	90	Fredrikstadt	540	Palermo	2359
Yarmouth	100	Friederichshafen	512	Port Mahon	2077
Calmar	818	Ghent	215	Salerno	2449
Carlscrona	768	Gothenburg	528	Savona	2285
Copenhagen	630	Grimstadt	450	Syracuse	2479
Cronstadt	1300	Halmstadt	590	Tarragona	1935
Dantzig	905	Hamburg	390	Tangiers	1425

The distances of various Ports from Hull in Nautical Miles—*continued.*

Ports.	Distance.	Ports.	Distance.	Ports.	Distance.
Tunis	2245	Tripoli	2524	Bushire	6,566
Valencia	1841	Venice	3106	Calcutta	7,897
		Volo	3077	Colombo	6,883
Alexandria	3245	Zante	2741	Karachi	6,264
Alexandretta	3505			Madras	7,157
Ancona	2996	Batoum	3817	Penang	8,164
Athens	2922	Constantinople	3232	Rangoon	8,034
Bari	2789	Galatz	3576		
Beyrout	3494	Ibrail	3588	Cape Town	6,282
Corfu	2738	Kertch	3663	Singapore	8,471
Corinth	2837	Kustenji	3430	Sydney	11,806
Fiume	3080	Odessa	3574		
Gallipoli	2665	Poti	3817	Bahia	4,639
Jaffa	3479	Sebastopol	3535	Buenos Ayres	6,533
Larnaca	3401	Sulina	3491	Colon (Panama)	5,000
Malta	2415	Taganrog	3821	Monte Video	6,423
Patras	2779	Trebizonde	3739	Santos	5,505
Port Said	3350	Varna	3381		
Ragusa	2843			Boston	2,913
Smyrna	3078	Aden	4774	New York	3,360
Trieste	3109	Bombay	6438	Quebec	3,093

The distances to various Ports from Leith in Nautical Miles.

Ports.	Distance.	Ports.	Distance.	Ports.	Distance.
Aberdeen	86	Plymouth	620	Bergen	360
Alloa	28	Southampton	510	Bremerhaven	440
Barry	826	Sunderland	125	Christiania	568
Cape Wrath	260	Tyne	118	Elsinore	596
Dublin (N.)	652	Yarmouth	300	Hamburg	490
Dundee	50				
Dover	374	Copenhagen	616	Boston, U.S.A.	2753
Falmouth	650	Cronstadt	1284	Halifax	2593
Grangemouth	21	Dantzig	875	New York	3180
Hartlepool	138	Riga	1067	Quebec	2933
Hull	240	Stockholm	1005	St John's(N.F.)	2105
Liverpool (N.)	686				
London	415	Antwerp	430		
Middlesbro'	148	Archangel	1691	Iceland	1050

The distances of various Ports from Hamburg in Nautical Miles.

Ports.	Distance.	Ports.	Distance.	Ports.	Distance.
Aberdeen	480	Alexandria	3,415	Odessa	3,745
Barrow	973	Aden	4,950		
Belfast	900	Bombay	6,620	Acapulco	12,580
Barry	813	Beyrout	3,665	Adelaide (S.C.)	11,170
Bristol	843	Cadiz	1,585	Auckland	12,250
Cardiff	821	Corunna	1,035	Callao	10,615
Cork (Queenstown)	811	Constantinople	3,405	Sydney, N.S.W.	11,980
		Colombo	7,070	Valparaiso	9,345
Cuxhaven	56	Calcutta	8,074		
Dover	392	Genoa	2,495	Buenos Ayres	6,710
Dublin	881	Gibraltar	1,630	Colon (Panama)	5,085
Dundee	495	Greytown	5,150	Monte Video	6,603
Falmouth	631	Havana	4,530	Pernambuco	4,435
Finisterre	1100	Hong Kong	10,185	Rio Janeiro	5,500
Glasgow	951	Jamaica	4,540	Santos	5,685
Grimsby	376	Lisbon	1,350		
Hartlepool	429	Malta	2,606	Baltimore	3,800
Havre	520	Marseilles	2,320	Boston	3,290
Hull	390	Naples	2,605	Charleston	3,970
Leith	490	New Orleans	5,060	Halifax, N.S.	2,950
Limerick	1001	Point de Galle	7,110	New York	3,510
Liverpool	964	Port Said	3,550	Quebec	3,310
London	434	Singapore	8,650	Sable Island	2,825
Manchester	1006	Tampico	5,350		
Milford Haven	781	Trinidad	4,280	Cape Town	6,460
Newport (Mon.)	833	Venice	3,276	Cape Verde Island	2,795
		Vera Cruz	5,340		
Plymouth	601			Delagoa Bay	7,610
Southampton	481	Batoum	3,990	Las Palmas	2,000
Sunderland	432	Galatz	3,747	Zanzibar (S.C.)	6,660

The distances of various Ports from Liverpool in Nautical Miles.

Ports.	Distance.	Ports.	Distance.	Ports.	Distance.
Belfast	150	Dover	591	Hull	760
Bristol	292	Dublin	126	Leith	686
Cardiff	270	Falmouth	320	Limerick	470
Cork	265	Finisterre	739	Lizard	323
Douglas (I.M.)	70	Galway	450	London	680
Drogheda	120	Glasgow	220	Londonderry	217

The distances of various Ports from Liverpool in Nautical Miles—*continued.*

Ports.	Distance.	Ports.	Distance.	Ports.	Distance.
Milford Haven	180	Brindisi	2,564	Cape Coast Castle	3893
Newry	130	Fiume	2,915	Cape Town	6135
Plymouth	352	Patras	2,608	Cape Verde	2470
Southampton	471	Port Said	3,208	Delagoa Bay	7283
St Catherine's (I. Wight)	478	Smyrna	2,913	Lagos	4283
Swansea	235	Trieste	2,944	Las Palmas	1665
Start Point	386	Batoum	3,652	Maderia	1435
Christiania	1005	Constantinople	3,067	Sierra Leone	3022
Copenhagen	1071	Odessa	3,405	St Helena	4560
Cronstadt	1739	Varna	3,216	Teneriffe	1660
Dantzig	1338	Aden	4,645	Zanzibar (S.C.)	6330
Kiel	1144	Calcutta	7,749	Bahia	4485
Stettin	1230	Hong Kong	9,853	Buenos Ayres	6386
Stockholm	1460	Point de Galle	6,786	Cape Horn	7355
Archangel	2058	Rangoon	7,890	Colon (Panama)	4548
Antwerp	704	Shanghai	10,603	Maranham	4035
Bergen	800	Singapore	8,323	Monte Video	6276
Bremerhaven	918	Yokohama (*v.* S.C.)	11,231	Para	4195
Gothenburg	974	Do. (*v.* P.C.)	8,050	Pernambuco	4085
Hamburg	964	Callao (*v.* P.C.)	5,930	Rio Janeiro	5175
Havre	517	Coquimbo	9,191	Santos	5345
Rotterdam	714	Honolulu (*v.* P.C.)	9,300	Demerara	3920
Bilbao	700	Iquique	9,775	Havana	4075
Bordeaux	677	San Francisco (*v.* P.C.)	7,836	Jamaica	4080
Brest	420	Valparaiso (*v.* P.C.)	7,202	New Orleans	4629
Cadiz	1230	Do. (*v.* C.H.)	9,006	Tampico	4916
Lisbon	1008	Vancouver I.	14,400	Vera Cruz	4910
Oporto	850	Wellington, N.Z. (*v.* C.H.)	11,605	Baltimore	3337
Ushant	390	Wellington, N.Z. (*v.* P.C.)	11,100	Boston, U.S.A.	2803
Vigo	800	Melbourne (S.C.)	11,172	Charleston	3485
Algiers	1700	Sydney (S.C.)	11,658	Halifax	2453
Almeria	1434	Azores	1,415	New York	3050
Barcelona	1801	Bonny	4,393	P. E. Island	2464
Genoa	2155			Philadelphia	3200
Gibraltar	1290			Portland (Me.)	2780
Marseilles	1975			Quebec	2803
Naples	2266			Sable Island	2321
Palermo	2194			St John's (N.F.)	1990
Venice	2941			St John (N.B.)	2670
Alexandria	3080				

The distances of various Ports from London in Nautical Miles.

Ports.	Distance.	Ports.	Distance.	Ports.	Distance.
Aberdeen	445	Boulogne	97	Aden	4,676
Ardrossan	730	Bremerhaven	390	Akyab	7,838
Barrow	675	Calais	96	Bombay	6,343
Barry	518	Cherbourg	236	Calcutta	7,799
Belfast	700	Dieppe	162	Colombo	6,793
Bristol	548	Dunkirk	114	Hong Kong	9,903
Cardiff	526	Elsinore	682	Karachi	6,166
Cork	540	Ghent	165	Madras	7,059
Dublin	600	Gothenburg	600	Muscat	5,894
Dundee	415	Hamburg	434	Nagasaki	10,853
Falmouth	345	Havre	215	Penang	8,066
Glasgow	775	Ostend	120	Point de Galle	6,836
Greenock	757	Rotterdam	183	Port Said	3,250
Hartlepool	300	St Malo	330	Rangoon	7,940
Hull	225			Saigon	8,993
Leith	415	Brest	420	Shanghai	10,653
Limerick	700	Lisbon	1041	Singapore	8,373
Liverpool	680	Vigo	855	Suez	3,340
Lynn	210			Yokohama (*v.* S.C.)	11,281
Milford Haven	476	Algiers	1750	Do. (*v.* P.C.)	8,100
Middlesbro'	295	Gibraltar	1326		
Newport(Mon.)	536	Marseilles	2040	Acapulco	12,308
Newry	630	Naples	2308	Auckland	11,970
Plymouth	325			Antofagasta	9,582
Sligo	800			Callao	10,340
Southampton	210	Alexandria	3130	Guayaquil	11,052
Sunderland	307	Athens	2807	Valparaiso	9,071
Swansea	506	Beyrout	3379	Wellington N.Z. (*v.* C.H.)	11,670
Tyne	315	Brindisi	2614	Wellington, N.Z. (*v.* P.C.)	11,150
Waterford	526	Jaffa	3364		
Yarmouth	135	Malta	2310	Adelaide (S C.)	10,897
		Port Said	3250	Auckland ,,	12,600
Christiania	700	Venice	2993	Batavia ,,	8,636
Copenhagen	702			Brisbane ,,	12,148
Cronstadt	1373	Constantinople	3117	Hobartown ,,	11,382
Helsingfors	1248			Manilla ,,	9,758
Riga	1153			Mauritius ,,	7,086
Stockholm	1090	Batoum	3702	Melbourne ,,	11,222
		Galatz	3461	Sydney ,,	11,708
Archangel	2050	Odessa	3459	Wellington ,,	12,612
Amsterdam	210	Sulina	3376		
Antwerp	180	Taganrog	3706		
Bergen	730	Varna	3266		

The distances of various Ports from London in Nautical Miles—*continued.*

Ports.	Distance.	Ports.	Distance.	Ports.	Distance.
Bahia	4530	Barbadoes	3810	Portland (Me.)	3003
Bermuda	3110	Belize	4740	Quebec	3030
Buenos Ayres	6438	Havana	4255	St John's (N.F.)	2202
Cape Horn	7458	Jamaica	4264		
Colon (Panama)	4700	Martinique	3785	Azores	1400
Monte Video	6328	Tampico	5076	Cape Town	6187
Para	4247			Cape Verde	2520
Rio Janeiro	5227	Boston, U.S.A.	3025	Delagoa Bay	7337
Rio Grande do Sul	5940	Halifax	2680	Lagos	4335
		New York	3260	Madeira	1480
Santos	5410	Philadelphia	3410	Teneriffe	1720

The distances of various Ports from New York in Nautical Miles.

Ports.	Distance.	Ports.	Distance.	Ports.	Distance.
Alexandria	8,050	Coquimbo	8,814	Naples	4,220
Amsterdam	3,325	Cronstadt	4,505	New Orleans	1,700
Antwerp	3,290	Cape Verde	2,890	Point de Galle	8,767
Azores	2,300	Demerara	2,200	(S.C.)	
Bahia	4,060	Galway	2,720	Para	2,900
Barbadoes	1,800	Gibraltar	3,250	Pensacola	1,630
Bermuda	690	Glasgow	3,120	Pernambuco	3,680
Bombay (S.C.)	8,251	Greytown	2,046	Port Said	5,180
Bordeaux	3,180	Havana	1,180	Reval	4,340
Buenos Ayres	5,750	Havre	3,170	Rio Janeiro	4,754
Calcutta (S.C.)	9,730	Hong Kong	11,830	Rio Grande	5,340
Caldera	9,080	(S.C.)		Rosario	5,980
Callao (*v.* P.C.)	3,362	Limon (Hon-	1,740	Santos	4,954
Aston (S.C.)	11,736	duras)		Shanghai (S.C.)	12,563
Almeria, Cape)	13,880	Lisbon	3,030	Singapore (S.C.)	10,300
Barcelona	6,820	Las Palmas	3,401	Stockholm	4,255
Genoa	7,000	Manilla (S.C)	11,625	Sydney (*v.* P.C.)	9,811
Gibraltar	1,900	Madras (S.C.)	8,981	San Francisco	5,645
Marseilles	628	Maranham	3,100	(*v.* Panama)	
Naples	2,074	Melbourne	12,260	Tampico	2,020
Palermo	2150	Monte Video	5,650	Trinidad	1,940
Venice	2941	Madeira	3,085	Valparaiso	8,646
Alexandria	3080			Do. (*v.* P.C.)	4,633

The distances of various Ports from New York in Nautical Miles—*continued.*

Ports.	Distance.	Ports.	Distance.	Ports.	Distance.
Vera Cruz	1,990	Cape Hatteras	340	Philadelphia	220
Vancouver (*v.* Panama)	6,350	Cape Race	1000	Portland (Me.)	410
		Charleston	628	Sandy Hook	18
Wellington (*v.* Horn)	10,800	Charlestown	425		
		Cape May (D.)	145	Punta Gorda	1200
Yokohama (*v.* Suez)	13,200	Cape Florida	1450	Quebec	1315
		Galveston	1850	Sable Island	685
Yokohama (*v.* Panama)	9,823	Halifax	580	Savannah	695
		Jamaica	1480	St Andrews (N.B.)	495
		Key West	1084		
Dundee	3,150	Martinique	1715	St John (N.B.)	520
Leith	3,180	Montreal	1450	StJohn's (N.F.)	1065
		Mobile	1660	Sydney (C.B.)	810
Bahamas	860	Nantucket Shoal	220	Tampa	1300
Baltimore	410			Trinidad	1940
Belle Isle	1,085	Nassau	960	Wilmington (N.C.)	495
Bermuda	690	Newport News	280		
Boston	330	Pensacola	1630	Washington	415

The distances of various Ports from Southampton in Nautical Miles.

Ports.	Distance.	Ports.	Distance.	Ports.	Distance.
Cardiff	338	Gothenburg	660	Alexandria	3193
Dundee	510	Hamburg	481	Athens	2618
Falmouth	160	Rotterdam	229	Brindisi	2425
Greenock	564	Bordeaux	520	Corfu	2434
Hartlepool	380	Cadiz	1105	Malta	2131
Hull	325	Corunna	555	Port Said	3069
Leith	510	Lisbon	869	Venice	2802
London	210	Vigo	660		
Plymouth	137			Constantinople	2928
Sunderland	393	Algiers	1561		
Yarmouth	220	Gibraltar	1153	Aden	4476
		Marseilles	1841	Bombay	6143
Bremerhaven	531	Messina	2167	Calcutta	7599
Christiania	725	Naples	2129	Colombo	6593
Elsinore	780	Palermo	2055	Hong Kong	9708

The distances of various Ports from Southampton in Nautical Miles—*continued.*

Ports.	Distance.	Ports	Distance.	Ports.	Distance.
Karachi	5,966	Cape Horn	7258	New York	3060
Madras	6,859	Colon (Panama)	4610	Philadelphia	3210
Nagasaki	10,653	Monte Video	6128	Portland	2803
Point de Galle	6,636	Pernambuco	3960	Quebec	2870
Port Said	3,069	Rio Janeiro	5027	Sable Island	2348
Shanghai	10,453	Santos	5210	St John's (N.F.)	2002
Singapore	8,173				
Suez	3,159	Barbadoes	3610	Azores	1290
Yokohama	11,081	Demerara	3880	Las Palmas	1523
Do. (*v.* P.C.)	7,950	Havana	4054		
Adelaide (S.C.)	10,697	Jamaica	4064	Cape Town	5987
Auckland	11,770	New Orleans	4589	Cape Verde	2320
(C.H.)		Tampico	4876	Delagoa Bay	7137
Brisbane (S.C.)	11,948	Trinidad	3810	Lagos	4135
Hobartown	11,182	Vera Cruz	4870	Madeira	1280
(S.C.)				Mauritius	8590
Melbourne	11,022	Boston, U.S.A.	2825	(S.C.)	
Sydney	11,508	Cape Race	2030	Sierra Leone	2874
Bermuda	2,910	Charleston	3495	St Helena	4412
Buenos Ayres	6,238	Halifax	2480	Zanzibar (S.C.)	6183

The distances of various Ports from Yokohama in Nautical Miles.

Ports.	Distance.	Ports.	Distance.	Ports.	Distance.
Batavia	3,163	New York (*v.* Panama)	9823	London (*v.* P.C.)	8,100
Bering Straits	2,300			Do. (S.C.)	11,281
Cape Flattery	4,163	San Francisco	4750	Liverpool (*v.* P.C.)	8,050
Hakodati	540	Valparaiso	9320	Do. (S.C.)	11,231
Hiogo	400			Southampton (S.C.)	11,081
Hong Kong	1,620				
Honolulu	3,440	Esquimalt (Vancouver)	4220	Do. (*v.* P.C.)	7,950
Kobe	345				
Nagasaki	681	Victoria (Vancouver)	4223		
Nagasaki (*v.* Inland Sea)	735				
		Vancouver (Race I.)	4213	Cardiff (*v.* S.C.)	11,094
New York (*v.* Suez)	13,630			Do. (*v.* P.C.)	7,963

The distances to various Ports from Hong Kong, Halifax (N.S.), and Cape Town in Nautical Miles.

Ports.	Distance.	Ports.	Dis. tance.	Ports.	Distance.
From Hong Kong *to*—		San Francisco	6,450	*From* Cape Town *to*—	
		Thursday Is.	2,763		
		Taku	1,510		
Amoor River	2,400	Tientsin	1,540	Cape Amber (Madagascar)	2300
Amoy	320	Vladivostock	1,680		
Bangkok	1,450	Valparaiso	10,525	Cape Coast Castlare	2600
Batavia	1,845	Woosung	808		
Bombay	3,930	Wei-hai-wei	1,250	Cape Gardafui	3750
Buenos Ayres	10,600	Yokohama	1,620	Cape St Mary (Madagascar)	1600
Bering Straits	4,000	Yangtse River Mouth	900		
Calcutta	3,094			Cape Verde Islands	3860
Canton	90				
Cape Town	7,000	*From* Halifax (N.S.) *to*—		Cape St Vincent (Portugal)	5180
Ceylon (Galle)	2,950				
Cheefoo	1,300			Colombo	4320
Foochow	460	Baltimore	885	Congo River	1800
Formosa	500	Barbadoes	1,900	Cook Town	7000
Honolulu	4,950	Bermuda	750	Cape Horn	3790
Haiphong	450	Boston	380	Callao	6661
Hang Chow Bay	800	Cape Race	455	Chinde (Zambesi)	1630
		Charleston	1,090		
Kobe	1,375	Demerara	2,280	Delagoa Bay	1150
Manilla	648	Guadaloupe	1,720	Durban	850
Melbourne	4,940	Gibraltar	2,500	East London	600
Nagasaki	1,067	Havana	1,650	Point de Galle	4300
New York (*v.* Panama)	11,297	Jamaica	1,800	Gibraltar	5130
		Liverpool	2,453	Hong Kong	7000
Nieuchwang	1,390	Montreal	880	Hobartown	6130
Ningpo	743	Nassau	1,345	Iquique	6150
Port Arthur	1,320	Newport News	755	Jamaica	6580
Peiho River	1,380	New York	580	Karachi	4630
Penang	1,804	Philadelphia	725	Lagos	2650
Port Darwin	2,371	Plymouth	2,355	Loanda	1650
River Min	460	Portland (Me.)	300	Madeira	4800
Rio Janeiro	10,270	Quebec	760	Madras	4892
Rangoon	2,580	Sable Island	170	Maulmain	5490
Saigon	916	Savannah	1,140	Mombasa	2480
Shanghai	900	St John's (N.F.)	520	Mossamedes	1275
Singapore	1,530	Sydney (C.B.)	250	Mozambique	2040
Sourabaya	1,923	Trinidad	2,050	Melbourne	6025
Swatow	190	Wellington (N.C.)	885	Muscat (P. Gulf)	4650
Sydney	4,400				

The distances to various Ports from Cape Town, Cardiff, Dover, Calcutta, and Bombay in Nautical Miles.

Ports.	Distance.	Ports.	Distance.	Ports.	Distance.
From **Cape Town**—*contd.*		Lizard Point	172	Cape Town	5430
		Longships	149	Chittagong	350
		Lundy Island	65	Colombo	1142
Manilla	6,500	Nash Point	20	Hong Kong	3094
New York	6,820	Start Point	239	Karachi	2360
Perth (W. A.)	4,900	St Catherine's Point	323	Madras	760
Port Darwin	6,255			Mauritius	3320
Port Elizabeth	450	Ushant	245	Maulmain	800
Port Nolloth	300			Penang	1300
Plymouth	5,897			Point de Galle	1210
Port Said	5,663	*From* **Dover** *to*—		Java	2596
Panama	6,840			Rangoon	780
Rio Janeiro	3,270	Calais	21·5	Singapore	1664
River Niger	2,500			Tamatave	3500
Rangoon	5,330	Antwerp	134	Zanzibar	3610
Sierra Leone	3,195	Amsterdam	172		
Singapore	5,560	Boulogne	25·5		
St Helena	1,800	Bremerhaven	380		
Sydney (N.S.W.)	6,480	Copenhagen	680	*From* **Bombay** *to*—	
		Dunkirk	37		
Sydney (C.B.)	7,540	Flushing	89	Aden	1637
San Francisco	10,140	Hamburg	392	Algoa Bay	4260
Tamatave (Madagascar)	2,450	Ostend	60	Batavia	2736
		Rotterdam	135	Bushire	1423
Vancouver	5,380	Schomven Lightship	91	Cape Town	4527
Walfish Bay	800			Colombo	940
Wellington	7,350	Skaw	538	Delagoa Bay	3560
Zanzibar	2,640	West Hinder Lightship	46	Point de Galle	980
				Karachi	500
From **Cardiff** *to*—		Ushant	298	Madras	1180
				Mauritius	2570
		From **Calcutta** *to*—		Rangoon	2060
Burlings	834			Seychelle Is.	1800
Dover	431			Singapore	2458
Europa Point	1,153	Aden	3269	Tamatave	2820
Finisterre	608	Akyab	375	Taticorin	780
Hartland Point	65	Bombay	1915	Zanzibar	2600

APPENDIX L.

The N.E. Coast Inst. E. and S. Guidance Specification for Triple Compound Engines of Cargo Steamers.

1. **Indicated Horse Power.**—For calculation purposes in this specification and in average sea conditions the I.H.P. is to be found as follows:—

$$\text{I.H.P.} = \frac{D^2SN}{700}$$

D = Diam. L.P. cylinder in inches.
S = Stroke in feet.
N = Revs. per minute. Found as per section 2.

Note.—The divisor is adjusted for a referred mean pressure of 30 lbs. per sq. in.

2. **Revolutions.**—$N = \frac{32(S+4)}{S}$.

3. **Boiler Pressure.**—180 lbs. per sq. in. (gauge).

4. **Ratios of Cylinder Areas.**—Ratio for 180 lbs. pressure.

H.P.	M.P.	L.P.
1	About 2·74	About 7·5
	1	,, 2·74

5. **Cuts-off at Sea Power.**—

About 57 per cent. 57 per cent. 55 per cent.

Each reversing lever to be fitted with expansion gear for adjusting the cuts-off in engines over 800 I.H.P.

6. **Speeds of Steam.**—The mean steam speeds to be calculated as follows:—

$$\frac{\text{Area of cylinder in sq. ins.} \times \text{piston speed in ft. per sec.}}{\text{Area of pipe, port, or opening in sq. ins.}} = \text{Speed in ft. per sec.}$$

Table of mean steam speeds in feet per second:—

	H.P.	M.P.	L.P.
Main steam pipe . . .	110	...	...
Port opening . . .	110	150	240
Steam ports . . .	80	85	100
Exhaust passage or pipe .	60	65	75

Steam Ports and Passages.—Width of ports to be about ·8 of diameter of cylinder; and the design of L.P. exhaust ports, passages, and pipe, to be such that the absolute pressure in L.P. cylinder, as shown by the exhaust line on indicator cards, does not differ by more than 1½ lbs. per square inch from that recorded in the steam space of the condenser at top.

7. **Maximum Load.**—The maximum load on main working parts to be taken as the product of the area of H.P. cylinder in inches and the boiler pressure in lbs. per sq. in. (gauge).

8. **Crank Shaft.**—The diameter of crank shaft in body to be a multiple of ¼ in., and not less than ⅛ in. above Lloyd's Rule, and the proportions of the remaining parts to be not less than the following:—

1. Diameter of crank pin to be equal to diameter of shaft + ½ in.
2. Diameter of crank shaft in web to be equal to diameter of shaft + ½ in.
3. Diameter of webs to be equal to diameter of crank pin by 1·85.
4. Thickness of webs to be equal to diameter of shaft by ·62.
5. Thickness of couplings to be equal to diameter of shaft by ·25.
6. Six coupling bolts to be used for shafts up to and including 15 ins. diameter. Nine coupling bolts to be used for shafts above 15 ins. diameter.

*7. Diameter of pitch circle of coupling bolts to be 1·48 diameter of crank shaft.

*8. Diameter of coupling bolts to be equal to—

$$\cdot 7 \times \sqrt{\frac{\text{diam. of shaft}^2 \text{ in inches}}{\text{number of bolts} \times \text{diam. of pitch circle in inches}}}.$$

Bolts to be parallel.

**Note.*—These two rules may be varied, provided that equivalent strength is given. Shaft to be in three interchangeable pieces.

9. **Length of Connecting Rods.**—Length of connecting rod between centres to be twice the stroke or four times the crank radius.

Diameter of Connecting Rod.—Diameter at centre of length to be same diameter as piston-rod body.

Connecting-rod Top Ends.—Connecting rods to have single top end gudgeons for all engines having H.P. cylinders of 24 ins. diameter and under.

10. **Crosshead Guides.**—Main crosshead guides to be of the single type in all sizes of engine.

Load on Main Crosshead Guides.—Maximum load in lbs. on crosshead guides to be taken as:—

$$\frac{\text{Area of H.P. cylinder in sq. ins.} \times \text{boiler pressure in lbs. per sq. in. (gauge)}}{4}.$$

11. **Maximum Pressures on Principal Bearing Surfaces.**

	Lbs. per sq. in.
Main bearings	250
Crank pins	500
Crosshead gudgeons	1000
Guide shoes (ahead)	55
Guide shoes (astern)	110

Diameter by length to be taken as area of bearings.
Overall length by overall breadth as area of guide shoes.

12. **Maximum Stresses on Principal Working Parts.**

	Lbs. per sq. in.
Ingot steel piston rod at screw	5500
Piston-rod body (after deducting ¼ in. from diameter to allow for returning)	3000
Piston and connecting-rod bolts at screw	5500
Main bearing bolts at screw	4500
Main bearing keeps (if forged)	6000
Connecting rod bottom end keep (if forged)	7500
Piston rod keep (if forged)	7500
Bolts at bottom end of valve spindle, at bottom ends of eccentric rods, and at eccentric straps	3500
Bolts at top ends of eccentric rods	3000
Screw at top end of valve spindle	2000
Pump levers—for design with double plates	4000
Pump levers—for design with single plate	3200

Notes.—The keeps are calculated as beams with distributed loads and supported ends.
Loads on valve gear bolts found as per section 13.
Load on pump levers found as per section 22.
Levers calculated as for bending stress only.
All dimensions to be to the nearest ¼ in. above the calculated size for dimensions of 2 in. and upwards, and to the nearest ⅛ in. above the calculated size for dimensions below 2 in.

13. **Valve Gear.**—The valve-gear proportions to be determined from the load on L.P. slide valve spindle calculated as follows:—

$$\text{Load in lbs.} = 4(L \times B).$$

Where L = Overall length of L.P. valve face in inches.
B = Overall breadth of L.P valve face in inches.

Valve Spindles.—Diameter at gland to be not less than $= \frac{\text{diam. piston rod at gland}}{2} + \frac{1}{2}$ in.

Eccentric Rods.—Eccentric rods to be parallel in the body and the diameter to be not less than $= \frac{\text{diam. piston rod at gland}}{2} + \frac{1}{4}$ in.

Reversing Shaft.—Diameter to be not less than = diameter piston rod at gland − 1 in.

Maximum Pressures on Bearing Surfaces of Valve Gears.

	Lbs. per sq. in
Link block gudgeons	500
Link block slippers	300
Eccentric rod top end pins	500
Eccentric sheaves (ahead and astern)—cast iron on cast iron	70
Eccentric sheaves (ahead and astern)—white metal on cast iron	90

Notes.—All eccentrics and straps to be duplicates.

Eccentrics to be on body of shaft and not on couplings.

All dimensions to be to the nearest $\frac{1}{4}$ in. above the size calculated.

14. **Thrust Block.**—When of horse-shoe type the pressure on thrust collars not to exceed 60 lbs. per square inch when calculated from indicated thrust, which is to be determined as follows:—

$$\text{Lbs. Indicated thrust} = \frac{\text{I.H.P.} \times 33{,}000}{\text{propeller pitch in ft.} \times \text{revs. per min.}}.$$

15. **Main Engine-driven Reciprocating Circulating Pump (Double-acting).**—The circulating pump to be proportioned to deliver an amount of condensing water equal to 40 times the feed water, with a displacement efficiency of ·8. The feed water to be taken at 15 lbs. per I.H.P. per hour.

16. **Maximum Speeds of Circulating Water.**—The speeds of circulating water are to be calculated as follows:—

$$\frac{\text{Area of bucket in sq. ins.} \times \text{bucket speed in ft. per sec.}}{\text{area of passage in sq. ins.}} = \text{speed of water.}$$

Approximate Speeds in Feet per Second.

	Feet per second.
Main injection	11
Passages in pump	6
Valve grids	7
Past life of valves	12
Discharge pipe	9

17. **Main Engine-driven Reciprocating Air Pump.**—Capacity of air pump not less than one-sixteenth of the capacity of L.P. cylinder.

18. **Condenser.**—When a condenser of the ordinary type is used, [illegible]10. Condensing surface to be 1½ square feet per I.H.P. type in all [illegible]denser to be so arranged that the vacuum in the air pump **Load on** [illegible] does not differ from that in the condenser steam space at crosshead guides [illegible]an two-tenths of an inch of mercury, or 2·72 inches of

Area of H. [illegible]nperature between the condenser steam space at top [illegible] as discharged by the air pump, and before admix- [illegible]ther source of heat, not to exceed 20° F.

of surface above specified includes an allowance

for the prejudicial effects upon condensing efficiency of oil and scale on the tubes, and of residual air in the condenser.

19. **Alternative Production of Vacuum by Steam Jet.**—When a steam jet is employed for the withdrawal of air, and the condenser is suitably proportioned and arranged, the circulating water to be 32 times the feed calculated as per section 15, the air pump to be one-twentieth of the capacity of L.P. cylinder, and the loss of temperature between the steam space in condenser at top and the condensate as discharged by the air pump, and before admixture with drains or other source of heat, not to exceed 10° F.

20. **Utilisation of Heat in Auxiliary Exhaust Steam.**—Provision to be made for securing the complete absorption by the feed water of the heat in the exhausts from the various auxiliaries, including the steering engine, electric-light engine, and the evaporator.

Note.—With the proportions already specified, and with a reasonably airtight system, a vacuum of at least 27 ins. will be maintained in the steam space at condenser top, at sea temperatures up to 70° F., and under these conditions the whole of the heat can, in ordinary circumstances, be utilised in the feed water.

21. **Main Engine-driven Feed Pumps.**—Capacity of *each* engine-driven feed pump 1/700th of capacity of L.P. cylinder.

22. **Pump Gear.**—Load on pump gear to be calculated as follows:—Load in lbs. = 25 (area of air-pump bucket + area of circulating-pump bucket) + 15 (area of both feed-pump rams + area of both bilge-pump rams). All in sq. ins.

Maximum Pressures on Pump-Gear Bearing Surfaces.—

	Lbs. per sq. in.
Pump link pins	400
Engine link pins	300
Pump lever centre gudgeon bearings . .	250

Note.—For cargo vessels of large power it is recommended that the circulating and feed pumps be independently driven pumps.

23. **Evaporator.**—An evaporator to be fitted capable of producing 10 tons of fresh water per 24 hours for each 1000 I.H.P. of main engines.

24. **Prevention of Delivery of Oil to Boilers.**—It is desirable that oil should be separated from the auxiliary exhaust steam before it is used to heat the feed water, but in any case the feed water to be passed through a satisfactory filter before delivery to boilers.

25. **De-aeration of Feed Water.**—With the object of preventing boiler corrosion, adequate means to be provided for the de-aeration of the feed water before it enters the boilers.

26. **Prevention of Heat-loss by Lagging.**—All surfaces radiating utilisable heat, including the feed pipes and auxiliary exhaust pipes, to be covered with an adequate thickness of non-conducting material.

Cargo Steamers Triple Compound Engines. N.E. Coast Inst. E. and S. Standard Rule: Revolutions = 32(S + 4) ÷ S. Revolutions and Piston Speeds by it.

Stroke.	Revolutions.	Piston Speed.	Stroke.	Revolutions.	Piston Speed.
feet.	per min.	ft. per min.	feet.	per min.	ft. per min.
1·50	117	352	3·25	72	464
1·75	105	368	3·50	69	480
2·00	96	384	3·75	66	496
2·25	89	400	4·00	64	512
2·50	83	416	4·25	62	528
2·75	78	432	4·50	60	544
3·00	75	448	4·75	59	560

Diameter of L.P. Cylinders of Triple Compound Engines by Rule: I.H.P. = $D^2 \times S \times N \div 700$.

I.H.P.	Stroke of Pistons in inches.													
	18	21	24	27	30	33	36	39	42	45	48	51	54	57
300	34·5	33·8	33·0	32·4	31·8	..	..	..	..	..	..	..	..	..
350	37·3	36·5	35·7	35·0	34·4	..	..	..	..	..	..	..	..	..
400	..	39·0	38·0	37·4	36·8	36·0	..	..	..	..	..	..	..	..
450	..	41·4	40·5	39·6	39·0	38·0	37 5	..	..	..	..	..	..	..
500	..	..	42·7	41·8	41·0	40·2	39·6	..	..	..	..	..	..	..
600	..	..	46·8	45·8	45·0	44·0	43·3	42·6	..	..	..	..	..	..
700	..	..	..	47·8	48·7	47·6	46·8	46·0	45·2	..	..	..	..	..
800	..	..	..	..	52·0	50·9	50·0	49·2	48·4	47·5	..	..	..	..
900	..	..	..	..	55·0	54·0	53·0	52·1	51·3	50·4	49·8	..	..	..
1000	..	..	..	..	..	56·9	56·0	55·0	54·1	53·1	52·5	..	..	..
1100	..	..	..	..	..	59·7	58·7	57·6	56·7	55·7	55·1	..	..	..
1200	..	..	..	..	..	62·4	61·3	60·2	59·2	58·2	57·5	56·5	..	..
1300	..	..	..	..	..	64·9	63·8	62·7	61·7	60·6	59·9	58·8	..	..
1400	..	..	..	..	..	..	66·3	65·0	64·0	63·0	62·1	61·0	..	..
1500	..	..	..	..	..	..	68·6	67·3	66·2	65·1	64·3	63·1	62·2	..
1600	..	..	..	..	..	..	..	69·5	68·4	67·2	66·4	65·2	64·2	..
1700	..	..	..	..	..	..	..	..	70·5	69·2	68·4	67 2	66·1	..
1800	..	..	..	..	..	..	..	..	72·5	71·2	70·4	69·2	68·0	67·1
1900	..	..	..	..	..	..	..	..	..	73·2	72·4	71·1	70·0	69·0
2000	..	..	..	..	..	..	..	..	..	75·0	74·2	72·9	71·7	70·7
2200	..	..	..	..	..	..	..	..	..	..	77·8	76·4	75·3	74·2
2400	..	..	..	..	..	..	..	..	..	..	81·3	79·9	78·6	77·5
2600	..	..	..	..	..	..	..	..	..	..	..	83·0	81·7	80·5
2800	..	..	..	..	..	..	..	..	..	..	..	..	84·9	83·7
3000	..	..	..	..	..	..	..	..	..	..	..	..	87·9	86·7
3 500	..	..	..	..	..	..	..	..	..	..	..	..	95·0	93·6
4 000	..	..	..	..	..	..	..	..	..	..	..	..	..	100

APPENDIX M.

Russian Weights and Measures with British and Metrical Equivalents.

Linear Measure.

1 vershok = 1·75 ins. = 45·5 millimetres.
1 arshine = 16 vershoks = 28 ins. = 711 ,,
1 sazhene = 3 arshines = 7 ft. = 2·135 metres.
1 verst = 500 sazhenes = 0·67 mile = 1073 ,,
1 foot = 6·86 vershoks = 0·43 arshine.
1 yard = 1·286 arshines = 0·4286 sazhene.
1 mile = 1·509 versts = 754·29 sazhenes.

Square Measure.

1 sq. arshine = 0·60 sq. yd. = 0·507 sq. metre.
1 sq. sazhene = 9 sq. arshines = 49 sq. ft. = 4·552 sq. metres.
1 sq. dessiatine = 2,400 sq. sazhenes = 2·7 acres = 10,928 ,,
1 sq. verst = 250,000 sq. sazhenes = 281·2 acres.
1 square yard = 1·667 square arshines.

Cubic Measure.

1 cub. vershok = 5·36 cub. ins. = 878 cub. cms.
1 cub. arshine = 4096 cub. vershoks = 12·7 cub. ft. = 0·3595 cub. mtr.
1 cub. sazhene = 9 cub. arshines = 12·7 cub. yds = 9·7 cub. mts.
1 cubic foot = 0·0787 cubic arshine = 322 cubic vershoks.

Money Values (Normal).

1 copeck = 0·0213 shilling = 0·255 penny = 2·68 centimes
100 copecks = 1 gold rouble = £0·1063 (2s. 1½d.) = 2·68 francs.
£1 sterling = 9·42 roubles. 1 shilling = 47·1 copecks.
1 penny = 3·925 copecks.

Weights.

1 dolya = 0·6856 grains.
1 zolotnik = 96 dolyas = 2·40 drams.
1 lot = 3 zolotnik = 0·4502 oz. avoir. = 0·0126 kilograms.
1 funt = 32 lots = 0·90 lb. avoir. = 0·41 ,,
1 pood = 40 funts = 36·11 ,, = 16·41 ,,
1 berkovetz = 10 poods = 3·214 cwts. = 164 ,,
1 British ton = 62 poods. 1 cwt. = 3·1 poods. 1 lb. avoir. = 35·5 lots.

Dry Measure.

1 garnet		= 2·88 quarts	= 3·275 litres.
1 chetverik =	9 garnets	= 5·77 gallons	= 26·248 ,,
1 osmina =	4 chetveriks	= 11·55 pecks	= 104·95 ,,
1 chetvert =	2 osminas	= 5·77 bushels	= 209 90 ,,
1 last =	12 chetverts	= 8·66 quarters	= 2419 ,,

1 quarter = 1·386 chetverts. 1 bushel = 1·385 chetveriks.

Liquid Measure.

1 chaska		= 0·216 pint	= 0·123 litre.
1 koushka =	10 chaskas	= 2·16 pints	= 1·23 litres.
1 shtoff =	12·5 chaskas	= 2·70 ,,	= 1·538 ,,
1 vedro	= 10 koushkas = 8 shtoffs	= 10·82 quarts	= 12·33 ,,
1 bochka =	40 vedros	= 108·3 gallons	= 492 ,,

1 gallon = 3·704 koushkas = 2·96 shtoffs. 1 pint = 4·63 chaskas.

APPENDIX N.

Hydraulic and Steam Tests of the Admiralty and Register Societies.

Tests and Trials.

Admiralty Hydraulic Tests are as follows:—

Air reserve bottles, &c., belong thereto	4000 lbs.
Boiler tubes and fuel heater tubes under 2 inches	2500 ,,
Solid-drawn boiler tubes above 2 inches	1500 ,,
Each condenser, distiller and oil-cooler tubes	1000 ,,
Feed-pump chambers, pipes, valves, &c.	3 times boiler pressure.
Ash ejector pump and pipes	600 lbs.
Main steam pipes, valves, &c. } Auxiliary pipes, valves, &c. }	twice the boiler pressure.
Boiler-room oil-feed pumps, &c.	400 lbs.
Boiler in shop and on board	$1\frac{1}{2}$ × W.P.
Cylinders, &c., of auxiliary engines	$1\frac{1}{2}$ × W.P.
Suction passage, &c., of feed pumps	300 lbs.
Low-pressure cylinders of auxiliary engines	235 ,,
Pump ends, discharge valves, &c., fire and bilge pumps	250 ,,
High-pressure ahead turbine, steam end and cover	255 ,,
High-pressure ahead, remainder	200 ,,
After completion of this turbine	170 ,,
Cruising turbine and astern turbine	170 ,,

Low-Pressure Turbine:—

Astern cylinders and inlet covers	50 lbs.
Ahead cylinders and inlet covers	30 ,,
Exhaust chamber and passages	30 ,,
On completion, the whole	30 ,,
Receiver pipe and valve between cruising and high-pressure ahead turbines	255 ,,
Feed suction pipes	250 ,,
Pump chambers of distilling, forced lubrication, engine-room oil-feed pumps and lime tanks	100 ,,
All underwater valves not over 12 inches diameter	100 ,,
Receiver pipes between high-pressure ahead and low-pressure turbines	60 ,,
Air pumps, auxiliary exhausts, evaporator cases, bilge suctions, sluice valves, oil-fuel suctions	60 ,,
Receiver pipes, high-pressure astern and low-pressure astern turbines	50 ,,

Working sides of underwater valves over 12 inches diameter	50 lbs.
Air-pump pipes and connections	50 ,,
Main eduction pipes	30 ,,
Steam chambers of condensers, auxiliary engine drain tanks, steam chambers and distillers	30 ,,
Shaft easings before and after fitting	30 ,,
Water chambers of condensers, circulating pumps, cases, valves, &c. &c.	25 ,,
The feed and oil tanks	10 ,,
Boiler 0·44 ultimate strength of boiler, &c., or	W.P. + 90 ,,

Valves and valve boxes to be tested with valves open and with valves shut, and afterwards tested by steam.

Steam tests are also to be applied to all high-pressure pistons and slide valves of auxiliary machinery; all reducing valves both before and after being fitted in the ship.

The Italian Government require the following tests:—

High-pressure cylinder and valve chests when W.P. is less than 142 lbs. to 1·5 W.P., when over (W.P. + 71) lbs.	
Second cylinder and valve chests to	W.P.
Third cylinder and valve chests to	0·66 ,,
Fourth cylinder and valve chests to	0·33 ,,
Steam jackets of all cylinders	2 × W.P. in them.
High-pressure casing of a turbine	1·33 × W.P.
Astern-going casing of a turbine	W.P.
Low-pressure casing of a turbine, admission end	0·33 ,,
Low-pressure casing of a turbine, exhaust end	28½ lbs.
Condensers when built up and all condensers for turbines	28½ ,,
Cast-iron condensers for reciprocators	21 ,,
Safety valves and feed pumps	2 × W.P.
Air and circulating pumps	28½ lbs. per sq. in.

British Corporation require as follows:—

Hydraulic tests of boilers	1·5 × W.P. + 50 lbs.
Steam pipes of copper	2 × W.P.
Iron or steel	3 × W.P.
Feed pipes	2·4 W.P. when copper.
Steel	3·6 × W.P.

Lloyd's Register require as follows:—

Hydraulic tests of boilers	1·5 × W.P. + 50 lbs.
Lap-welded boiler tubes, 2 per cent. to	750 lbs. per sq. in.
Seamless steel boiler tubes all to	1000 ,, ,,

Copper tubes, test pressure $= 4500\frac{t}{D}$.

(t is the thickness, and D is the internal diameter in inches.)
Condenser tubes all to 300 lbs. per sq. in.

Steel and iron lap-welded steam pipes	3 × W.P.
Air receivers (Diesel engines)	2 × W.P.

Board of Trade require as follows:—

Hydraulic tests of boilers	1·5 × W.P. + 50 lbs.
High-pressure turbine cases to	1·33 × W.P.
Low-pressure condenser end	30 lbs. per sq. in.
Astern-going case	W.P.

Admiralty Steam Trials are as follows:—

1. 30 hours' duration trial at ⅓rd H.P.
2. 30 hours, the first 8 at ⅘ths H.P., and 22, remainder, at ⅔ths H.P.
3. Full H.P. for 8 consecutive hours, with waste of water not exceeding 3½ tons per 1000 H.P. per 24 hours.
4. Various short trials to test handling of engines, speed going astern, both with all and half the boilers, &c. &c., the testing of auxiliaries, &c. &c.
5. The engines are then opened up for examination, &c.
6. Final trial of 24 hours, half the time at half H.P., the remainder as may be directed, all the boilers being with steam up.

Bureau Veritas (*vide* pp. 641–42) require the following tests:—

Compound engine cylinders:

(*a*) High pressure to working pressure	+ 85 lbs.
(*b*) Medium pressure, triples to	0·8 W.P.
(*c*) First medium pressure, quadruples to	W.P.
(*d*) Second medium pressure, quadruples to	0·5 W.P.
(*e*) Low pressure of all to	43 lbs.
Turbines, high pressure casings, and astern-going H.P. ends	1·25 W.P.
Boilers to 2 W.P.; above 142 lbs.	W.P. to W.P. + 142 lbs. On board, to W.P. + 85 lbs.
Steam and feed pipes to 2 W.P.	Condensers, 28·5 lbs.

British Marine Engineering Design and Construction Committee:—

Hydraulic tests of new boilers	1·5 × W.P. + 50 lbs.
Boilers which have been on service	1·5 × W.P.
Boiler smoke tubes	1000 lbs. per sq. in.
Stop and safety valve chests and boiler mountings generally to	2 × W.P.
Copper tubes to a pressure which will produce a stress of	7500 lbs.
Copper steam pipes to	2 × W.P.
Copper feed pipes (delivery)	2·5 × W.P.
Wrought-iron or steel steam pipes	3 × W.P.
Wrought-iron or steel feed pipes	4 × W.P.
Cast-iron or steel pipes generally	3 × W.P.

APPENDIX O.

MELTING POINTS OF VARIOUS METALS AS ASCERTAINED BY PROFESSOR CARNELLY, IN DEGREES CENTIGRADE.

Silver	between 916° and 954°.	Commercial, 960° and 1034°.	Pure, 1040°.
Aluminium	,, 700 ,, 850.		
Antimony	,, 425 ,, 432.	,, 432° ,, 440°.	
Bismuth	,, 256 ,, 260.	,, 260° ,, 270·5°.	
Cadmium	,, 307 ,, 326.		
Cobalt	,, 1371 ,, 1800.		
Copper	,, 950 ,, 1100.	,, 1100° ,, 1236°.	,, 1330° and 1398°.
Iron, white cast	,, 1075 ,, 1250.	Grey cast, 1200° ,, 1600°.	Ordinary cast, 1530°.
Iron, wrought-bar	,, 1500 ,, 1600.	Pure bar, 1600°.	Pure iron, 1587°. Steel, 1375°.
Iridium	,, 2200 ,, 2500.		
Lead	,, 320 ,, 334.	Pure lead, 330° and 335°.	
Magnesium	,, 433 ,, 500.		
Manganese	,, 1482 ,, 1900.		
Nickel	,, 1371 ,, 1600.		
Platinum	,, 1700 ,, 2534.		
Tin	230.		
Zinc	between 342 and 412.	Commercial, 414° and 450°.	Boils at 940°.

Alloys	Cu 75 + Ag 25	melts at 1114°;	Cu 50 + Ag 50	melts at 941°;	
,,	Pb 82 + Sb 18	,, 260°;	Pb 90 + Sb 10	,, 240°;	
,,	Pb 25 + Sn 1	,, 558°;	Pb 10 + Sn 1	,, 541°;	
,,	Pb 1 + Sn 1	,, 370°;	Pb 1 + Sn 5	,, 378°;	
,,	Pb 30·5 + Zn 69·5	,, 190°;	Pb 50 + Zn 50	,, 202°;	
,,	Sn 6 + Zn 1	,, 204°;			
,, Britannia metal	Sb 18 + Zn 82	,, 204°.			

APPENDIX P.

Lloyd's Register of Shipping.

New Unified Rules for the Survey and Construction of Engines and Boilers of Steam Vessels.

Section 1.—In steam vessels, the machinery and boilers are to be inspected throughout, the boilers tested by hydraulic pressure, and the machinery tested under steam by the Society's Engineer-Surveyors, who will furnish a report to the Committee. If found satisfactory, the Committee will thereupon grant a certificate, and insert in the Register Book the notification, "L.M.C." *in red* (*i.e.* "Lloyd's Machinery Certificate"), indicating that the machinery and boilers are certified to be in good order and safe working condition.

Section 2.—1. In steam vessels built under Special Survey, the machinery and boilers must also be constructed under Special Survey.

2. In cases of machinery or new boilers being built under Special Survey, the distinguishing mark ✠ will be noted in red, thus: "✠L.M.C.," or "✠N.E. & B.," or "✠N.B."

3. In order to facilitate this inspection, the plans of the machinery and boilers are to be examined, and from them the working pressure fixed.

4. Any novelty in the construction of the machinery or boilers is to be reported to the Committee.

5. The Surveyors are to examine the materials and workmanship from the commencement of the work until the final test of the machinery under steam; any defects, etc., to be pointed out as early as possible.

6. The Surveyors may also, if desired, compare the work as it progresses with the requirements of the specification agreed upon by the parties concerned, and certify to the conditions thereof, as far as can be seen, being satisfactorily complied with.

Section 3.—1. All steam vessels must have at least two entirely separate means of supplying the main boilers with feed water, when working at full power.

2. If the main feed supply is from pumps worked from the main engines, then with engines above 70 N.H.P. there must be two main feed pumps, so arranged that either can be shut off and overhauled while the other is at work. Each pump is to be capable of performing the whole work required.

3. In engines of 70 N.H.P. and under, and also in all engines of

steam fishing vessels and of tugs and tenders, one main feed pump worked from the main engines will be approved, provided the auxiliary feed pump required by clause 5 is fitted.

4. The main feed pumps may be worked by independent engines, provided they are fitted with automatic regulators for controlling their speed. If only one such pump is fitted for the main feed, the auxiliary feed pump required by clause 5 should also be fitted with an automatic speed regulator.

5. A steam pump is to be provided as an auxiliary feed supply, capable of performing the whole work required of feeding the boilers at full power. This pump is to have suctions to the hotwell and to the sea. This pump may be also used for general purposes, but in this case the suction pipes to the hotwell and sea must be entirely distinct from those to the bilges.

6. Each feed pump, main or auxiliary, is to be fitted with a relief valve.

Section 4.—1. The engines are to be fitted with two bilge pumps so arranged that one can be overhauled whilst the other is at work.

2. In engines of 70 N.H.P. and under, and also in engines of steam fishing vessels, and of tugs and tenders, one bilge pump worked from the main engine will be sufficient, provided the steam pump required by clause 4 is fitted.

3. In lieu of bilge pumps worked from the main engines, a separate steam pump may be fitted additional to that required by clause 4.

4. A steam pump is to be provided connected to the main bilge system as required in clause 5.

5. The bilge pumps referred to in clauses 1, 2, 3, and 4 are to be arranged to pump for each compartment of the vessel except the peaks, and to deliver overboard. At least one of the steam pumps (clauses 3, 4) is to be fitted with a direct suction to the engine-room, which can be used while the other bilge pumps are being used on other parts of the vessel (see Section 39, clause 4). All bilge suction pipes are to be fitted with strum boxes or strainers, so constructed that they can be cleared without breaking the joints of the suction pipes. The total area of the perforations in the strainers should be not less than double that of the cross section of the suction pipe. The mud boxes and roses in engine-room are to be placed where they are easily accessible, and to the satisfaction of the Surveyor.

6. A bilge suction to the circulating pump is to be fitted.

7. One of the steam pumps referred to in clauses 3, 4, must be fitted to draw from the sea and to deliver water on deck.

Section 5.—In vessels fitted with twin screw engines, the main feed pumps and the main engine bilge pumps may be fitted, one of each on each engine, provided they are so connected that either feed pump can deal with the water from both hotwells, and that either bilge pump can draw from all parts of the vessel.

Section 6.—1. If an independent circulating pump is not fitted, one of the steam pumps referred to in Section 4, clauses 3, 4, must be arranged to circulate water through the condenser.

2. All discharge pipes are to have discharge valves fitted on the plating of the vessel in accessible positions, and if possible these are to be above the deep load line. The discharge valves are preferably to be of non-return type.

3. No pipes are to be carried through the bunkers without being properly protected.

Section 7.—1. In all steam pipes provision is to be made for expansion and contraction to take place without unduly straining the pipes. Where the provision for expansion is by means of bends in the pipes it is recommended that the various lengths of pipe should be made short of the designed lengths by amounts equal to half the calculated expansion at the temperature of the steam. Where the expansion is provided for by sleeves and packed glands, these should be of brass or gun-metal, means being provided to prevent the sleeves being drawn out when under pressure.

2. All steam pipes and also the exhaust pipes of all auxiliary engines should be provided with suitable drain cocks and pipes.

3. The exhausts from the steam steering engine and from the electric light engine are to be led by separate pipes and valves to the main engine condenser, or to an auxiliary condenser, or to the atmosphere where no auxiliary condenser is fitted.

4. For the thicknesses and tests of copper, iron, or steel steam and other pipes see Section 17.

Section 8.—In single screw vessels fitted with steam turbine engines arrangements shall be made so that steam can be led direct to the L.P. turbine and either the H.P. or I.P. turbine can exhaust direct to the condenser. Two turbines are to be fitted with astern wheels.

Section 9.—In all vessels fitted with steam engines in which the lubricating oil is circulated under pressure, a spare oil pump is to be supplied with all connections ready for immediate use, and two independent means are to be arranged for circulating water through the oil cooler.

Section 10.—Where the engines, thrust blocks, etc., are fitted directly upon the tank tops, the holes for the holding-down bolts are to be tapped and the bolts made watertight. In these cases a cast-iron chock is to be fitted between the tank top and the engine bed plate in way of each holding-down bolt.

Section 11.—A steam gauge attached to the main steam pipe and one to each steam receiver, and a vacuum gauge attached to the condenser are to be fitted in convenient places where they can be seen from the engine-room platform.

Section 12.—Gauges of an approved description for testing the truth of the crank shafts of reciprocating engines are to be supplied with all new engines, and adjusted in the presence of the Surveyor. Where steam turbine engines are fitted, gauges for testing the truth of the rotor, pinion, and gear wheel shafts are to be supplied, and adjusted in the presence of the Surveyor.

Section 13.—Where feed heaters or evaporators are fitted, the parts subjected to pressure if made of mild steel are to be constructed to

meet the requirements of the rules for boilers, etc. (Sections 27 to 35). Where pressure parts are made of cast iron, the portions in direct tension are to be made 3½ times the thickness which would be required if they were made of mild steel, and those parts which are subjected to cross breaking stress only are to be made of twice the thickness which would be required if they were made of mild steel. Similar parts made of brass or gun-metal should be 2½ and 1½ times respectively the thickness required in mild steel. These parts are to be tested by hydraulic pressure to twice the working pressure coming upon them.

Cocks, Pipes, Sea Connections, and Pumping Arrangements.

Section 14.—1. All sea cocks and valves required to be fitted on the plating of the vessel are to be fixed in easily accessible places above the level of the stokehold and engine-room platforms, or are to be fitted on strong iron or steel stands of a height sufficient to lift them up to the level of these platforms.

2. The bolts securing all cocks or sea connections to the plating of the vessel are to be tapped into the plating of the vessel or fitted with countersunk heads.

3. The blow-off cocks on the plating of the vessel are to be fitted with spigots passing through the plating, and a brass or gun-metal ring on the outside. The cocks are to be so constructed that the key or spanner can only be taken off when the cock is shut. (See Section 23, clause 3.)

4. Cocks and valves connecting all bilge suction pipes are to be fixed in accessible places above the stokehold and engine-room platforms. They should be marked to indicate their purpose. If bilge suction valves are placed in the shaft tunnel, they must be made workable from a position above the bulkhead deck.

5. All valve and valve seats in connection with bilge suctions are to be of brass or gun-metal. The valves should close with a right-hand motion.

6. The arrangements of pumps, bilge suction and delivery pipes are to be such as will not permit of water being run from the sea into the vessel by an act of carelessness or neglect.

Rules for Shafts.

Material.

Section 15.—1. Shafts for marine engines may be forged from ingot steel, wrought-iron bars cross piled, scrap wrought iron, or from mild steel scrap, and they may be made by hammer, press, or by rolling mill.

2. If scrap is used, it must be selected of practically uniform quality and be thoroughly cleaned. A mixture of iron and steel scrap is not permitted in any case, and steel scrap is not to be used for screw shafts, thrust shafts, or for crank shafts.

3. When ingot steel is used it must comply with the conditions of Section 39.

4. The webs of built-up crank shafts may be made of ingot steel, scrap wrought iron, or of scrap mild steel, forged, rolled, or pressed. They may also be made of steel cast to shape, with eye-holes suitable for boring to size to fit the journals and pins. If made of cast steel, the material should have a tensile strength not exceeding 32 tons per square inch, and the sum of the tensile strength in tons per square inch and the elongation per cent. measured on a standard test piece shall be not less than 50. (See Section 38, clause 6.)

5. Couplings of wrought-iron or scrap-steel shafts may be welded to the body. Those of ingot-steel shafts may be forged from the solid or formed by upsetting the shaft ends by hydraulic pressure. Couplings may also be made separately of ingot steel, wrought iron, or scrap mild steel forged, or they may be steel castings. In such couplings provision must be made for the astern pull, either by checked rings or other approved means.

6. In ingot steel shafts made by rolling and in which the couplings are formed from the shaft by upsetting, the test pieces are to be cut from the outer edges of the couplings.

7. All shafts are to be turned all over and are to be examined when rough turned and also when finished.

Rules for determining the Sizes of Shafts.

Section 16.—1. In steam turbine engines the diameters of intermediate shafts are not to be less than those given by the following formula :—

$$\text{Diameter of intermediate shaft in inches} = \sqrt[3]{\frac{SHP}{R}} \times F,$$

where SHP is the maximum designed horse-power to be transmitted through the shaft when working at the maximum power;

R is the number of revolutions per minute when working at the maximum power;

F = 64 for ocean-going and cross-channel ships;
= 58 for vessels working in rivers and smooth waters.

2. The diameter of the wheel shaft of a geared turbine engine, where there are two pinions geared into it, opposite or nearly opposite to one another, should not be less than 1·05 times that required for the intermediate shaft. Where there is only one pinion gearing into the wheel, or where there are two set at an angle less than 120°, the diameter of the shaft at the wheel and the adjacent journal shall not be less than 1·1 times that required for the intermediate shaft. Abaft the journal the diameter may be tapered to 1·05 times that required for the intermediate shaft.

3. The diameters of intermediate shafts for reciprocating steam engines shall not be less than those given by the following formula :—

$$\text{Diameter of intermediate shaft} = \sqrt[3]{\frac{D^2 \times S \times P}{c(r+2)}},$$

where D is the diameter in inches of the low-pressure cylinder, or the equivalent diameter where two or more low-pressure cylinders are used;

S is the stroke of piston, in inches;

P is the working pressure in the boilers, in lbs. per square inch;

r is the ratio of the swept volumes of the low-pressure and of the high-pressure cylinders;

c is a coefficient as per the following table :—

General Description of Engines.	Service of Ship.	
	Ocean-going and Cross-channel.	Rivers and Smooth Water.
(*a*) 2 cranks at 90°, cylinders compound	1800	2000
(*b*) 1 crank or 2 cranks at 180°, cylinders compound	1260	1400
(*c*) 3 cranks at 120°, cylinders compound, triple, or quadruple .	2000	2250
(*d*) 4 cranks balanced, cylinders compound, triple, or quadruple .	2000	2250
(*e*) 4 cranks at 90°, cylinders quadruple	1900	2150

4. Crank shafts for reciprocating engines shall have a diameter not less than 1·05 times that required for the intermediate shafts.

5. The diameter between the collars of thrust shafts transmitting torque shall not be less than 1·05 times that required for the intermediate shafts. This may be tapered down outside the collars to the size required for the intermediate shafts.

6. In twin or multiple screw ships, where intermediate shafts pass through stern tubes, the diameter of the tube shafts shall in no part be less than 1·05 times that required for the intermediate shafts, and any part of them in the tube exposed, or likely to be exposed, to sea water, shall not be less than 1·075 times that required for the intermediate shafts.

7. Tail shafts (shafts carrying the screw propellers) within the stern bush, whether within the stern tube or in the bracket, shall have a diameter not less than that given by the following formula :—

$$\text{Diameter of tail shaft under the liners, in inches} = D + \frac{P}{C},$$

where D is the diameter, in inches, required for the intermediate shaft;
P is the diameter, in inches, of the propeller;
C is a coefficient as given in the following table:—

	Values of C.	
	Where Shafts are fitted with Continuous Liners.	Where Shafts are not fitted with Continuous Liners.
For shafts for turbine engines . .	144	120
For shafts for reciprocating engines for ocean and cross-channel services .	120	100
For shafts for reciprocating engines for river and smooth-water services .	144	120

8. Tail shafts which are in the stern tubes should have the end forward of the stern gland tapered down to 1·05 times the diameter of the intermediate shafts.

9. In fast-running machinery, in order to prevent intermediate shafts from bending to such an extent as to cause whipping, the distance between the edges of adjacent bearings should not exceed that given by the following formulæ:—

Greatest length between adjacent bearings in feet

$$= C\sqrt{\frac{D}{R}} \text{ in the cases of solid shafts,}$$

or

$$= C\sqrt[4]{\frac{D^2+d^2}{R^2}} \text{ in the cases of hollow shafts;}$$

where D is the outside diameter of the shafts, in inches;
d is the diameter of the hole in the cases of hollow shafts, in inches;
R is the maximum number of revolutions of the shaft per minute;
C is 125 for outboard shafts;
145 for inboard intermediate shafts;
175 for intermediate shafts in ships engaged exclusively in smooth-water services.

10. In solid couplings the thickness of the flanges at the coupling-bolt holes shall not be less than the diameter required for the coupling bolts. For tail shafts the thickness of the coupling shall not be less than one-quarter of the diameter required for the intermediate shaft.

11. The radius of the fillet connecting the coupling flange to the

body of the shaft shall not be less than 0·125 times the diameter of the shaft.

12. The diameter of the coupling bolts shall not be less than given by the following formula:—

$$\left.\begin{array}{l}\text{Diameter of coupling bolts in inches,}\\ \text{measured at the face of the coupling}\end{array}\right\} = \sqrt{\frac{D^3}{3{\cdot}5n \times r}},$$

where D is the diameter required for the intermediate shaft, in inches;
n is the number of bolts in the coupling;
r is the distance of the centre of the bolt holes from the centre of the shaft, in inches.

13. The diameter over the screw-thread of the coupling-bolts should be at least $\frac{1}{4}$ inch less than that of the body of bolts which do not exceed $1\frac{3}{4}$ inches in diameter, at least $\frac{1}{4}$ inch less for bolts over $1\frac{3}{4}$ inches and up to 3 inches in diameter, and may be $\frac{1}{2}$ inch less in bolts over 3 inches in diameter. The points of the bolts should be reduced below the thread and made taper for a length of $\frac{3}{8}$ of the diameter of the bolt, so that the bolts may be driven out without damaging the thread.

14. In built crank shafts the width of the web, measured parallel to the axis of the shaft, shall not be less than 0·625 times the diameter required for the crank shaft, and the thickness of the metal round the eyes should be such that $h \times t^2$ is not less than 0·12 times D^3. That is:—

h should not be less than $0{\cdot}625 \times D$,
$h \times t^2$ should not be less than $0{\cdot}12\ D^3$,

where D is the diameter of the crank shaft, in inches;
h is the width of the crank web, in inches;
t is the thickness of metal round the eye measured radially, in inches.

15. Crank webs should be shrunk on the journals and crankpins, or be forced on by hydraulic pressure. They should be fitted with one or two keys or cylindrical dowels at the junction of the journals and webs. If only one dowel is fitted, its diameter should be 0·2 times the diameter of the shaft.

16. The liners to tail and tube shafts, if fitted, must be made of hard tough brass or bronze, shrunk on or forced on the shaft by hydraulic pressure, and no pins to secure them shall be fitted. Each liner must be cast in one piece; or, if made in two or more pieces, the junctions are to be made by electric welding or by fusing through the whole thickness of the liners. Joints made by butting, overlapping, stepping, or by soldering with soft solder are not permitted.

17. The thickness, in 64ths of an inch, of the liners of tail or tube shafts at the bearing portions should be not less than $13\sqrt{D}$, where D is the diameter of the shaft, in inches.

For the part of continuous liners between the bearing portions the thickness should be reduced to $10\sqrt{D}$.

18. The diameter of the screw for the nut at the end of the tail shaft should not be less than 0·7 times the diameter of the shaft under the liner. The depth of the nut should not be less than 0·625 the diameter of the shaft. The nut must be provided with efficient stops to prevent it from unscrewing.

19. The length of the aftermost bearing or stern bush shall not be less than four times the diameter required for the tail shaft.

20. The after end of the liner upon the tail shaft must be fitted watertight into the propellor boss.

21. Where hollow shafts are used, the diameter in inches of an equivalent solid shaft is $\sqrt[3]{\frac{D^4 - d^4}{D}}$, where D is the outside diameter and d is the diameter of the hole in the hollow shaft, both in inches.

Steam and Other Pipes.

Section 17.—1. No pipe made from the electro deposition of copper on a mandril shall be used for steam, feed delivery, blow-off, or scum.

2. All copper steam, feed, blow-off, and scum pipes must be properly annealed before fixing in place.

3. All copper pipes used for steam, feed, blow-off, and scum purposes subject to a pressure over 75 lbs. per square inch shall be solid drawn.

4. No steam pipe intended for a working pressure over 180 lbs. per square inch shall be of copper when the internal diameter exceeds 5 inches. No copper pipe shall be used for superheated steam.

5. All copper steam pipes on completion and prior to being fitted in place shall be subjected to an hydraulic test of at least twice the working pressure to which they will be subjected.

6. All copper feed delivery pipes shall be hydraulically tested to at least 2½ times the working pressure of the boilers.

7. The working pressure to be allowed on copper pipes is to be determined by the following formula :—

$$WP = \frac{(t-3) \times F}{D},$$

where WP is the working pressure, in lbs. per square inch ;
D is the internal diameter, in inches ;
t is the thickness, in 100ths of an inch ;
F for solid-drawn steam pipes is 60 ;
for solid-drawn feed pipes is 48 ;
for brazed steam pipes 45 ;
for brazed feed pipes 36.

8. When copper pipes are bent, they must be made thicker than required by clause 7 to provide for the thinning at the bend. In no case should the radius of curvature at the centre line of the pipe be less than twice the external diameter of the pipe.

9. Steam and other pipes may be made of wrought iron or of wrought steel.

10. The process of welding the seams of iron or steel pipes shall be such that it is done by hammering or rolling the joint.

11. On completion of any work on iron or steel steam pipes which involves heating, whether for welding the joint, welding on flanges, bending the pipe, or for any other purposes, the pipe should be carefully annealed.

12. Mild steel for lap-welded or riveted steam pipes may have a tensile strength not exceeding 28 tons per square inch, with a minimum elongation of 25 per cent. on a standard test piece with a gauge length of 8 inches.

13. Feed pipes if made of steel should be solid drawn, cold finished.

14. All iron or steel steam pipes prior to being fitted in place shall be subjected to an hydraulic test of at least three times the working pressure to which they will be subjected.

15. All iron or steel feed delivery pipes shall be hydraulically tested to at least four times the working pressure of the boilers.

16. The working pressure to be allowed upon iron or steel steam and feed pipes shall be determined from the following formulæ:—

a. For solid-drawn cold finished steel steam pipes,

$$WP = \frac{(t-10)}{D} \times 120 ;$$

b. For solid-drawn hot finished steel steam pipes,

$$WP = \frac{(t-12)}{D} \times 120 ;$$

c. For lap-welded steam pipes of iron or steel whether with or without covering straps,

$$WP = \frac{(t-12)}{D} \times 90 ;$$

d. For feed delivery pipes,

$$WP = \frac{(t-8)}{D} \times 100 ;$$

where WP is the working pressure to which the steam pipes will be subjected, and in the case of feed delivery pipes is the boiler pressure;
D is the internal diameter of the pipes, in inches;
t is the thickness, in 100ths of an inch.

Boilers.

Section 18.—1. In the design and construction of marine boilers the following conditions should be observed:—

2. All steel plates, rivets, and bars used in the construction of boilers must be tested and found to conform to the requirements of the Rules (Sections 36, 37).

3. No steel plates subject to a direct tensile stress are to be welded except where the weld is covered by a butt strap or straps. For small steam domes, by special permission, where the welding is done by hammer, and the plates do not exceed ½ inch in thickness, the straps may be omitted. The strength shall in such cases be assumed to be 50 per cent. of that of the solid plate.

4. All steel plates which are welded, dished, flanged, or locally heated are to be afterwards efficiently annealed.

5. Butt straps must be cut from plates and not from rolled strips.

6. All rivet holes must be drilled, and as far as possible they should be drilled in place. After drilling the plates the burrs should be removed and the faying surfaces of the plates cleaned, and the sharp outer edges of holes removed also.

7. No steel stays are to be welded. If plus threads are desired, the ends of the stay bars may be upset or the bars may be drawn down in the central portions from bars originally of the size of the ends. In either of these two cases the bars must be subsequently annealed throughout. In double-ended boilers the through longitudinal stays must be supported at or near the middle of their length.

8. Screw stays of combustion chambers when fitted with nuts should be, as far as possible, normal to the chamber plates. When this is not possible, they must be fitted with taper washers to provide a fair bed for the nuts.

9. Nuts to screw stays in combustion chambers shall be not less than ⅝ inch thick for stays up to 1½ inches diameter over threads, ¾ inch thick for 1⅝ and 1¾ inch stays, 1 inch thick for 1⅞ and 2 inch stays, and 1¼ inches thick for stays over 2 inches in diameter. The nuts shall be made of solid mild steel or of iron which must be without weld. The nuts for longitudinal stays shall be to the British standards appropriate to the diameters of the stays, the outside nuts having the thickness therein provided for ordinary nuts, and the inside nuts having the thickness provided for lock nuts.

10. All longitudinal stays and screw stays should have threads in accordance with British standard specification, and true to pitch, viz.:—

After June 30, 1921, all screw stays 1¼ inches in diameter and upwards should have 9 threads per inch, and all stays 2 inches in diameter and above passing through plates, and secured by nuts on each side of the plate, should have not more than 6 threads per inch.

It is desirable for the threads of all screw stays to be turned off between the parts fitting into the plates. It is also desirable that the outer ends of screw stays should have a hole $\frac{3}{16}$ inch diameter drilled axially to a distance ½ inch beyond the inner face of shell and end plates.

11. When jointed longitudinal stays are fitted between the front and back tube plates it is desirable that they should be fitted with pins having an effective sectional area 25 per cent. in excess of that of the stay. The pins may be slack in the holes, the total slackness being not more than $\frac{1}{16}$ inch. The pins must be as close as possible to the shoulder of the eye forging. The shoulder of the forging should

have a diameter not less than 1 inch greater than the diameter of the hole.

Section 19.—1. No boiler, whether main or auxiliary, in which oil fuel is used, shall have a damper or other possible means of obstruction to draught in the uptake or funnel.

2. Manholes in cylindrical shells must have their shorter axes arranged longitudinally. Where the cylindrical shell is cut for a manhole, compensation must be provided and must be such that the strength in way of the hole is not less than that required for the longitudinal joint.

3. The end plates in the steam space in way of uptakes are to be shielded from contact with the heated gases.

Section 20.—1. When a flat plate is flanged to stiffen it at a manhole or sighthole, to permit the same working pressure as would be allowed upon an unpierced plate, the depth of the flange measured from the outer surface is to be at least equal to $\sqrt{t \times w}$, where t is the thickness of the plate, in inches, and w is the minor axis of the hole, in inches.

2. The doors to manholes, mudholes, and sightholes must be built up or pressed to shape and annealed, or made from one thickness of plate with a machined recess for the jointing material. Their spigot part or the recess must not have a greater clearance than $\frac{1}{16}$ inch all round, *i.e.* the axes must not be less than $\frac{1}{8}$ inch smaller than the holes in which they are fitted.

3. The studs for securing all doors should be screwed through the plate, and be fitted with nuts on the inside, or bolts may be used screwed through the plate with the heads inside.

4. All boilers should have, when possible, means for ingress to permit of examination and cleaning of the inner surfaces of plates and tubes exposed to flame. When the boilers are too small to permit of this, there must be sightholes and mudholes sufficiently large and numerous to permit of the inside being satisfactorily cleaned.

5. When the cross tubes of vertical boilers are large there must be a sighthole in the shell opposite to one end of each tube sufficiently large to permit the tube to be examined and cleaned. These sightholes must be in positions accessible for that purpose.

Section 21.—In all new boilers working at pressures up to 100 lbs. per square inch the hydraulic test must be twice the working pressure. For boilers working at pressures greater than 100 lbs. per square inch the hydraulic test pressure must be $1\frac{1}{2}$ times the working pressure plus 50 lbs. per square inch.

Section 22.—1. Every boiler must have at least two independent means of indicating the water level in it, and have marked on it in a contiguous position, easily seen, the level of the highest part of the combustion chamber. One of these means must be a glass gauge or an equivalent. The other may be a set of test cocks.

2. Test cocks, where practicable, should be fitted direct on the boiler shell. A set must consist of at least three cocks except in boilers 7 feet 6 inches in diameter and under, when there may be two.

For vertical boilers above 7 feet high the set of test cocks must consist of at least three.

3. All single-ended boilers over 16 feet mean diameter shall have a glass water gauge on each side. Single-ended boilers under 16 feet in diameter shall have one glass water gauge near one side and one set of test cocks near the other side.

4. All double-ended boilers shall have a glass water gauge near each end on opposite sides, and a set of test cocks at each end.

5. The cocks of all water gauges must be accessible from positions free from danger in the event of the glass breaking.

6. If the water gauges are not fitted directly to the shell of the boiler, but to stand pillars or columns, it is desirable that these pillars or columns should be bolted directly to the shell of the boiler. If they are connected to the boiler by means of pipes, the pipes must have terminal cocks, not valves, fitted direct to the boiler shell. For boilers exceeding 10 feet in diameter the pillars shall not be less than $2\frac{1}{2}$ inches, and the connecting pipes not less than $1\frac{1}{2}$ inches internal diameter. For boilers exceeding 7 feet 6 inches but not exceeding 10 feet in diameter, the pillars shall not be less than 2 inches, and the pipes not less than $1\frac{1}{4}$ inches internal diameter; and for boilers 7 feet 6 inches in diameter and under, the pillars shall not be less than $1\frac{3}{4}$ inches, and the pipes not less than 1 inch internal diameter. The upper ends of the connecting pipes must be arranged so that there is no pocket or bend where an accumulation of water from the condensation of the steam can lodge. They should not pass through the uptake if they can be otherwise arranged. If, however, this condition cannot be complied with, they may pass through it by means of a passage at least 2 inches clear of the pipe all round, open for ventilation.

7. A salinometer cock or valve must be fitted direct to each boiler in a convenient position. It must not be on the water gauge standpipe.

8. Each boiler shall have a separate steam-pressure gauge. Double-ended boilers shall have a pressure gauge at each end. The gauges are to be placed where they are easily seen.

9. Each boiler must have at least two independent means of feed, each with its own check valve. In vertical boilers one of the means of feed may be an injector.

Section 23.—1. Each boiler must have a blow-off valve fitted direct to the shell. The valve and its connections to the sea need not be more than $1\frac{1}{2}$ inches in diameter, and may be generally about $\frac{1}{10}$ inch in diameter for each foot in diameter of the boiler. It should, however, be not less than $\frac{3}{4}$ inch in diameter.

2. Blow-off valves and scum valves (when these latter are fitted) of two or more boilers may be connected to one common discharge, but when thus arranged there must be screw-down non-return valves fitted for each boiler to prevent the possibility of the contents of one boiler passing to another.

3. The blow-off cock or valve on the ship's side must be fitted above the level of the stokehold plates, in an accessible position, and must

be arranged so that it can be readily seen whether it is open or shut. The cock handle must not be capable of being removed unless it is shut, and if a valve is fitted the wheel must be fixed to the spindle. The cock or valve must be fitted with a spigot passing through the plating and a brass or gun-metal ring on the outside.

Section 24.—One main stop valve must be fitted to each boiler direct on the shell of the boiler. There shall be as few auxiliary stop valves as possible so as to avoid piercing the boiler shell more than is absolutely necessary. The arrangement, however, must be such that when more than one boiler is fitted it is possible to supply the steam whistle, the steam steering gear, and the electric light machinery from at least two boilers.

Section 25.—1. At least two safety-valves must be fitted to each boiler. They must be arranged so that the springs and valves are cased in, that the valves cannot be overloaded when steam is up, that they can be lifted by easing gear, and turned round on their seats by hand, and in case of fracture of springs they cannot lift out of their seats. Easing gear must be arranged to lift all the safety-valves on a boiler together, and must be workable from some accessible place, free from steam danger.

2. Vertical boilers having more than 100 square feet of total heating surface must have two safety-valves each not less than 1·5 inches diameter; those having less than 100 square feet may have one valve not less than 2 inches diameter.

3. All the safety-valves of each boiler may be fitted in one chest, which must be separate from any other valve chest and must be connected direct to the boiler by a strong and stiff neck, the passage through which should have a cross-sectional area at least equal to one-half the aggregate area of the safety-valves in the chest. Each safety-valve chest shall have a means of draining it; the drain pipe shall lead to the bilge or tank clear of the boiler.

4. The minimum aggregate area of the safety-valves of the ordinary type in each boiler, whether coal-fired or oil-fired and whether working under natural, forced, or induced draught, shall be found by the following formula:—

$$\text{Aggregate area of safety-valves in square inches} = \text{Total heating surface of boilers in square feet} \times \left(\frac{C}{p+15}\right),$$

where p is the working pressure, in lbs. per square inch. C being 1·25 for boilers using coal, and 1·5 for those using oil fuel, and for all boilers with closed stokehold forced draught.

5. The waste-steam pipe and the passages leading to it should have a cross-sectional area not less in square inches than 0·01 times the total heating surface of the boiler in square feet, nor in any case should it be less than 1·1 times the combined areas of the safety-valves as given by the above rule.

6. All safety-valves must be set to the required pressure under

steam. During a test of 15 minutes with the stop valves closed and under full firing conditions the accumulation of pressure must not exceed 10 per cent. of the loaded pressure. During this test no more feed-water should be supplied than is necessary to maintain a safe-working water level.

Section 26.—1. All boiler mounting valves over 1½ inches diameter must have outside screws, and all are to be arranged to be shut with a right-hand motion of the wheels, and must have means for clearly indicating whether they are open or shut. The covers must be secured by bolts or studs.

2. All cocks and valves connected to the boiler shall be such that it is seen without difficulty whether they are open or shut. When boiler mountings are secured by studs, the studs must have a full thread holding in the plate for a length of at least one diameter. If the stud hole penetrates the whole thickness of the plate, the stud must be screwed right through the plate and be fitted with a nut inside the boiler. Where bolts are used for securing mountings, they must be screwed right through the plate with their heads inside the boiler.

3. Where a superheater is fitted which can be shut off from the boiler it must have a separate safety-valve fitted with easing gear. The valve as regards construction must comply with the regulations for ordinary safety-valves, but the easing gear may be fitted to be workable from the stokehold only. The superheater must also be fitted with a drain cock or valve to free it from water when necessary.

4. All stop and safety-valve chests and steam-pipe fittings when subjected to saturated steam only may be made of cast iron, but such valve chests and steam-pipe fittings when subjected to steam of a temperature above 425 degrees Fahr., must be of cast steel or other approved material.

Rules for Determining the Working Pressure to be allowed in New Boilers.

Cylindrical Shells.

Section 27.—1. For the cylindrical shells of steel marine boilers the maximum working pressure (which is designated by WP, and is in lbs. per square inch) to be allowed shall be calculated from the following formulæ:—

Where the thickness of the shell plates does not exceed 1¾ inches,

$$WP = \frac{(t-2) \times S \times J}{C \times D}.$$

Where the thickness of the shell plates exceeds 1¾ inches, and the longitudinal seams are made with double butt straps,

$$WP = \frac{(t \times S \times J)}{2{\cdot}85 \times D}.$$

In the above formulæ

t is the thickness of the shell plate, in 32nds of an inch;
S is the minimum tensile strength of the steel shell plates, in tons per square inch;
J is the percentage of strength of the longitudinal seams calculated by the methods described below;
C is a coefficient, which is 2·75 when the longitudinal seams are made with double butt straps, 2·83 when the longitudinal seams are made with lap joints and are treble-riveted, 2·9 when they are made with lap joints and are double-riveted, and 3·3 when they are made with lap joints and are single-riveted;
D is the inside diameter of the outer strake of plating of the cylindrical shell measured in inches.

2. The percentage of strength of a riveted joint (J) is found from the following formulæ (i.), (ii.), (iii.): (i.) and (ii.) are applicable to any type of joint, (iii.) is applicable only to that type of joint in which the number of rivets in the inner rows is double that in the outer row. The lowest value given by the application of these formulæ is to be taken as the percentage of strength of the joint.

(i.) Percentage of strength of plate at joint as compared with solid plate

$$= \frac{100(p-d)}{p};$$

(ii.) Percentage of strength of rivets as compared with solid plate

$$= \frac{100(S_2 \times a \times n \times C)}{S_1 \times p \times T};$$

(iii.) Percentage of combined strength of the plate at the inner row of rivet holes and of the rivets in the outer row

$$= \frac{100(p-2d)}{p} + \frac{100(S_2 \times a \times C)}{S_1 \times p \times T};$$

where p = pitch of rivets at outer rows, in inches;
d = diameter of rivet holes, in inches;
a = sectional area of one rivet, in square inches;
n = number of rivets which are fitted in the pitch p;
T = thickness of plate, in inches;
C = 1·0 for rivets in single shear as in lap joints;
C = 1·875 for rivets in double shear as in double butt-strapped joints;
S_1 = minimum tensile strength of plates, in tons per square inch;
S_2 = shearing strength of rivets, which is taken generally to be 23 tons per square inch, and may be 85 per cent. of the minimum tensile strength of the rivet bars.

3. Where the longitudinal seams are fitted with double butt-strapped joints, the outer butt strap should have at least 0·625 of the strength of the plate, and should be of sufficient thickness to permit of efficient caulking of its outer edges. The inner butt strap should be $\frac{1}{32}$ inch thicker than this.

In cases where the number of rivets in the inner rows is double the number in the outer row, this will require the thickness of the outer strap to be

$$t_o = \frac{5 \times (p-d)}{8 \times (p-2d)} \times T,$$

and that of the inner strap to be at least

$$t_n = \frac{5 \times (p-d)}{8 \times (p-2d)} \times T + \tfrac{1}{32} \text{ inch.}$$

4. In all cases the clear space between a rivet hole and the edge of a plate should not be less than the diameter of the rivet hole, *i.e.* the centre of the rivet hole should be at least $1\frac{1}{2}$ diameters distant from the edge of the plate.

In joints whether lapped or fitted with butt straps, in which there are more than one row of rivets and in which there is an equal number of rivets in each row, the distance between the rows of rivets should be not less than $0{\cdot}33p + 0{\cdot}67d$ with zigzag riveting, or $2d$ with chain riveting.

In joints in which the number of rivets in the outer rows is one-half of the number in each of the inner rows, and in which the inner rows are chain riveted, the distance between outer rows and the next rows should be not less than $0{\cdot}33p + 0{\cdot}67d$ or $2d$, whichever is the greater, and the distance between the rows in which there are the full number of rivets should be not less than $2d$.

In joints in which the number of rivets in the outer rows is one-half of the number in each of the inner rows, and in which the inner rows are zigzag, the distance between the outer rows and the next rows should be not less than $0{\cdot}2p + 1{\cdot}15d$, and the distance between the rows in which there are the full number of rivets should be not less than $0{\cdot}165p + 0{\cdot}67d$.

In the above p is the pitch of the rivets in the outer rows.

5. The maximum pitch of the rivets in the longitudinal joints of boiler shells is to be—

$$\text{Maximum pitch in inches} = C \times T + 1\tfrac{5}{8} \text{ inches,}$$

where T is the thickness of the plate, in inches, and C is a coefficient as given in the following table:—

Number of Rivets per Pitch.	Coefficients for Lap Joints.	Coefficients for Double Butt-strapped Joint.
1	1·31	1·75
2	2·62	3·50
3	3·47	4·63
4	4·14	5·52
5	...	6·00

6. If holes are cut in the cylindrical shells of boilers for fixing of mountings, the diameters of the holes being greater than $2\frac{1}{2}$ times the thickness of the shell plating plus $2\frac{3}{4}$ inches, compensation must be fitted as in the case of manholes. (See Section 19, clause 2.)

7. When more than three screw stays pierce the cylindrical shell in a horizontal line, if d is their diameter and p the pitch, $\frac{100(p-d)}{p}$ should not be less than the percentage of strength required for the shell longitudinal joints. If this is not possible, the stays must be arranged out of line with one another longitudinally.

8. The riveting of the seams joining the end plates to the cylindrical shell shall be not less than 42 per cent. of that of the solid shell plate. Where the shell plates exceed $\frac{5}{8}$ inch in thickness, the seams connecting the shell plates to the end plates are to be double-riveted. Where the shell plates exceed $\frac{1}{2}$ inch in thickness, the intermediate circumferential seams of double-ended boilers are to be at least double-riveted.

9. The circumferential seam at or near the middle of the length of single-ended boilers shall have a strength of joint not less than 60 per cent. of the solid plate. The inner circumferential seams of double-ended boilers shall have a strength of joint not less than 62 per cent. of the solid plate. In any case there shall be three rows of rivets when single-ended boilers have shell plates over $1\frac{3}{8}$ inch in thickness and when double-ended boilers have shell plates over $1\frac{5}{16}$ inch in thickness.

10. The circumferential seams of the shells of vertical boilers shall have a strength of not less than 42 per cent. of the solid plate. When these seams are not complete circles, and when the shell plates exceed $\frac{5}{8}$ inch in thickness, the riveting shall be double.

Furnaces.

Section 28.—1. The working pressure to be allowed on corrugated furnaces is to be determined by the following formula:—

$$WP = \frac{C(t-1)}{D},$$

where D is the external diameter measured at the bottom of the corrugations, in inches;

t is the thickness of the furnace plate, in 32nds of an inch, measured at the bottom of the corrugation or camber;

C is a coefficient which is 480 for the Fox, Morison, Deighton, Purves, and other similar furnaces, and is 510 for the Leeds Forge Bulb Suspension furnace.

2. The working pressure to be allowed on plain furnaces or furnaces strengthened by the Adamson or other joints, and on the cylindrical bottoms of combustion chambers, is to be determined by the following formulæ, the least pressure obtained by either formula being taken:—

$$WP = \frac{C(t-1)^2}{(L+24) \times D}$$

or

$$WP = \frac{C_1}{D} \times [10(t-1) - L],$$

where D is the external diameter of the furnace or combustion chamber, in inches;

t is the thickness of the furnace plate, in 32nds of an inch;

L is the length of the furnace or of combustion chamber bottom or the length between points of substantial support, in inches, measured from the centres of rivet rows or from the commencement of flange curvature, whichever is applicable;

C is 1450 where the longitudinal seams are welded, and 1300 where they are riveted;

C_1 is 50 where the longitudinal seams are welded, and 45 where they are riveted;

WP is the working pressure, in lbs. per square inch.

3. When the furnaces are tapered, the diameter to be taken for calculation purposes shall be the mean of that at the top and of that at the bottom where it meets the substantial support from flange or ring. The length for the same purpose shall be the distance from the centre of the row of rivets connecting the crown to the body of the furnace to the substantial support at the bottom of the furnace, or to a row of screwed stays connecting the furnace to the shell, provided the pitch of stays at the furnace does not exceed 14 times the thickness of the furnace plate when the stays are riveted at their ends, and 16 times when they are fitted with nuts. Such screwed stays must be in diameter over the threads not less than 2·25 times the thickness of the furnace plate.

4. Where the furnaces are spherical in form and convex upwards at their tops, and are without support from stays of any kind,

$$WP = \frac{275(t-1)}{R},$$

where t is the thickness of the top plate, in 32nds of an inch;

R is the outer radius of curvature of the furnace, in inches.

5. For the ogee ring which connects the bottom of the furnace to the shell, and sustains the whole load on the furnace vertically,

$$WP = \frac{140(t-1)^2}{D \times (D-d)},$$

where t is the thickness of the ogee ring, in 32nds of an inch;
D is the inside diameter of the boiler shell, in inches;
d is the outside diameter of the lower part of the furnace where it joins the ogee line.

6. No furnace, plain or corrugated, should exceed $\frac{13}{16}$ inch in thickness.

Flat Plates supported by Stays secured in Various Ways.

Section 29.—1. The working pressure to be allowed on flat plates supported by stays is to be calculated by the following formula:—

$$WP = \frac{(t-1)^2 \times C}{a^2 + b^2}.$$

In this formula and in that in paragraphs 8 to 13,
WP is the working pressure, in lbs. per square inch;
t is the thickness of the flat plate, in 32nds of an inch;
t_w is the thickness of the washers, strips, or doublings employed, also in 32nds of an inch;
a is the distance apart of the rows of stays, in inches;
b is the pitch of the stays in the rows, in inches;
C is a coefficient which varies with the method of fixing the stays as follows:—

Where the plates are exposed to flame and the stays are screwed into the plate and the ends are riveted over, C=50. Where the plates are not exposed to flame and the stays are screwed into the plate and their ends are riveted over, C=57. In these cases the thickness of the plate must be at least half the diameter of the stay required by the Rules.

Where stay tubes are screwed into tube plates and expanded, C=52. If they are fitted with nuts, C=72.

Where the plates are exposed to flame and the stays are screwed into the plate and fitted with nuts on the outside, C=75. Where the plates are not exposed to flame, C=86.

Where the stays pass through the plates not exposed to flame and are fitted with nuts inside and outside, C=96.

2. Where plates are stiffened by flanging, the inner radius of which is not greater than $2\frac{1}{2}$ times the thickness of the plate, for the support thus given, C=110 when the plates are not exposed to flame, and C=96 when they are exposed to flame. The pitch is to be reckoned from the commencement of the curvature.

3. For portions of plate where the stays are irregularly pitched, d^2 is to be used instead of $a^2 + b^2$, d being the diameter of the largest circle

which can be drawn passing through not less than three points of support—viz. the centres of stays or the commencement of the curvature of flanging, whichever is applicable. In this case C is to be taken as the mean of the values appropriate for the points of support.

4. For the tops and sides of combustion chambers the distance between the rows of stays nearest to the back tube plate or the back plate respectively and the commencement of curvature of these plates at their flanges shall not be greater than a.

5. It is desirable that the stays of the combustion chambers should be so placed that the seams of the plates can be caulked without removing the stay nuts.

6. For the tops of combustion chambers where they are joined to the sides by curved portions, if the outer radius of the curved portion is less than half the allowable distance between the girders, the distance between the first girder and the inner surface of the side plate should not exceed the allowable distance between the girders. If the radius of the curved portion is greater than half the allowable distance between the girders, the width of the flat portion measured from the centre of the girder should not be more than half the allowable distance between the girders.

7. Where portions of plate are supported by stays secured in different ways, the value of C to be taken is the mean of the values appropriate to the method of securing the supporting stays.

8. Where the plates are supported by stays passing through them and are fitted with nuts inside and washers and nuts outside, the diameter of the washers being at least 3·5 times that of the stay, and their thickness at least two-thirds of that of the plate, but not greater than that of the plate, the working pressure may be

$$\mathrm{WP}=\frac{100}{a^2+b^2}[(t-1)^2+0{\cdot}15t_w^2].$$

9. Where the washers have a diameter of at least two-thirds of the pitch of the stays and a thickness of at least two-thirds of the thickness of the plate, but not greater than that of the plate, and are riveted to the plate in an efficient manner:—

$$\mathrm{WP}=\frac{100}{a^2+b^2}[(t-1)^2+0{\cdot}35t_w^2].$$

10. Where the plate is stiffened by strips at least two-thirds of the pitch of the stays in breadth and have a thickness at least two-thirds of that of the plate, but not greater than that of the plate, and are riveted to the plate in an efficient manner:—

$$\mathrm{WP}=\frac{100}{a^2+b^2}[(t-1)^2+0{\cdot}55t_w^2].$$

11. Where the plates are fitted with doubling plates having a

thickness of at least two-thirds of that of the plate, but not greater than that of the plate, and are riveted to them :—

$$WP = \frac{100}{a^2 + b^2}[(t-1)^2 + 0{\cdot}85t_w^2].$$

12. For the portions of tube plates in the nests of tubes,

$$WP = \frac{C(t-1)^2}{p^2},$$

where t is the thickness of the tube plate, in 32nds of an inch ;
p is the mean pitch of the stay tubes supporting any portions of the plate (being the sum of the four sides of the quadrilateral divided by 4) ;
C = 38 when the stay tubes are screwed and expanded into the plate and no nuts are fitted ;
C = 49 when the stay tubes are screwed and expanded into the plates and fitted with nuts.

13. For the wide water spaces of front tube plates between the nests of tubes and between the wing rows of tubes and the shell,

$$WP = \frac{C[(t-1)^2 + 0{\cdot}55t_w^2]}{a^2 + b^2},$$

where t is the thickness of the front tube plate, in 32nds of an inch ;
t_w is the thickness of the doubling plate, when so fitted, in 32nds ;
a is the horizontal pitch of stay tubes, in inches, measured across the wide water space ;
b is the vertical pitch of stay tubes in the bounding rows, in inches, measured from centre to centre ;
C = 52 when the stay tubes are screwed and expanded into the tube plates, and no nuts are fitted ;
C = 72 when the stay tubes are screwed and expanded into the tube plates, and nuts are fitted to each stay tube ;
C = 63 when the stay tubes are screwed and expanded into the tube plates, and nuts are fitted only to alternate stay tubes.

14. If steel of less strength than 26 tons per square inch is used for flat plates, the working pressure allowed shall be correspondingly reduced.

Section 30.—The pressure to be allowed on tube plates shall be calculated by the following formulâ, in which the compressive stress is taken at 14,000 lbs. per square inch :—

$$WP = 875 \times \frac{(D-d) \times t}{W \times D},$$

where t is the thickness of the tube plate, in 32nds inch ;
D is the horizontal distance apart of the tubes, centre to centre, in inches ;
d is the internal diameter of the plain tubes ;
W is the width of combustion chamber measured inside from tube plate to back chamber plate, or between tube plates in double-ended boilers with combustion chambers common to two opposite furnaces.

Section 31.—When vertical boilers have a nest or nests of horizontal tubes so that there is direct tension on the tube plates due to the vertical load on the boiler ends or to their acting as horizontal ties across the shell, the thickness of the tube plates and the spacing of the tubes must be such that the section of metal taking the load is sufficient to keep the stress within that allowed on the shell plates.

Further, each alternate tube in the outer vertical rows of tubes must be a stay tube. The tube plates between the stay tubes must be in accordance with the rules for tube plates as in Section 29, clause 12, and, in addition,

$$WP = \frac{(t-2) \times S \times (p-d) \times 100}{2{\cdot}9 \times D \times p},$$

where S is the minimum tensile strength of the steel plate, in tons per square inch ;
t is the thickness of the tube plate, in 32nds of an inch ;
D is twice the radial distance of the centre of the outer row of tube holes from the axis of the shell, in inches ;
p is the vertical pitch of tubes ;
d is the diameter of the tube holes, in inches.

Girders.

Section 32.—For girders supporting the tops of combustion chambers the following formula is to be used :—

$$WP = \frac{C \times d^2 \times t}{(L-P) \times D \times L} \times \frac{S}{28},$$

where d is the depth of the girder at centres, in inches ;
t is the thickness of the girder at centre, when this is a forging, or the sum of the thicknesses of the plates when the girder is made of two plates, measured in 32nds inch ;
L is the length of the girder in inches, measured between the tube plate and back chamber plate inside, or between tube plates in chambers common to two opposite furnaces ;
P is the pitch of stays supported by the girder, in inches ;
D is the distance apart of the girders, centre to centre, in inches ;

S is the minimum tensile strength of the steel plates forming the girder, in tons per square inch. In the case of forged girders S is to be taken as 24 for iron and 28 for steel;
C is a coefficient as follows:—

$$C=\frac{n}{n+1}\times 495,$$

when the number of stays in each girder is odd;

$$=\frac{n+1}{n+2}\times 495,$$

when the number of stays in each girder is even, n being the number of stays to each girder.

Stays.

Section 33.—1. For screw stays with threads not coarser than 9 threads per inch, made of steel or of special wrought iron tested to the requirements of the Rules (Sections 36, 37), the following formula is to be used, but in no case must the stress exceed 9000 lbs. per square inch:—

$$WP=\frac{(d-0{\cdot}267)^2\times 8250}{a},$$

where d is the diameter of the stay over the thread, in inches;
a is the area, in square inches, supported by one stay.

2. For steel longitudinal stays with threads not coarser than 6 threads per inch the working pressure is to be calculated from the following formula, but in no case must the stress exceed 11,000 lbs. per square inch when steel of a minimum tensile strength of 28 tons per square inch is used:—

$$WP=\frac{(d-0{\cdot}340)^2\times 9500}{a}\times\frac{S}{28},$$

where d is the diameter of the stay over the thread, in inches;
a is the area, in square inches, supported by one stay;
S is the minimum tensile strength of the steel, in tons per square inch.

3. In cases where longitudinal stays are made with enlarged ends and the body of the stay is smaller in diameter than at the bottom of the thread, and in cases where coarser threads than 6 per inch are used, the working pressure is to be calculated from the following formula:—

$$WP=\frac{(d^1-0{\cdot}125)^2\times 9500}{a}\times\frac{S}{28},$$

where d^1 is the diameter of the stay at the bottom of the thread or at the smallest part of the body.

Boiler Tubes, Plain and Stay

Section 34.—1. The following table gives the working pressures permissible with plain boiler tubes of standard thicknesses, whether of lapwelded wrought iron or of lapwelded or seamless mild steel:—

Outside Diameter of Tube in Inches.	Standard Thicknesses in L.S.G.				Working Pressures in lbs. per sq. inch.			
	A	B	C	D	A	B	C	D
2		11	10	9		155	215	300
2¼	11	10	9	8	140	190	260	315
2½	11	10	9	8	125	175	230	300
2¾	11	10	9	8	110	160	215	275
3	10	9	8	7	140	190	250	300
3¼	10	9	8	7	130	180	230	280
3½	10	9	8	7	120	165	215	260

2. On stay tubes, whether of wrought iron or of lapwelded steel, a working stress of 7500 lbs. per square inch of the net sectional area at the bottom of the thread is allowed.

3. The minimum thickness of stay tubes measured under the threads shall be ¼ inch for marginal stay tubes and $\frac{3}{16}$ inch for other stay tubes.

4. Stay tubes are to be screwed at both ends with continuous threads, and the holes in the tube plates are to be tapped with continuous threads. The thread should not be finer than 10 threads per inch. It is desirable, however, that they should be screwed to the standard 9 threads per inch, and after June 30, 1921, this should be the rule. The stay tubes are to be expanded by roller expanders and not made tight by caulking only.

5. No nuts are to be fitted to stay tubes at the combustion chamber end.

6. If stay tubes are required to have their thickness increased at the screwed ends so that the thickness at the bottom of thread is practically the same as in the body of the tube, the thickening is to be attained by upsetting and not by any welding process, and the tubes are to be annealed after the upsetting.

Section 35.—1. For ends of steam chests, crowns of vertical boilers, etc., dished outwards to partial spherical form and not fitted with stays, the following formula is to be used:—

$$WP = \frac{15 \times S(t-1)}{R},$$

WP is the working pressure, in lbs. per square inch ;
t the thickness, in 32nds of an inch ;
R the inner radius of curvature of the end in inches, which shall not exceed the diameter of shell ;
S the minimum tensile strength of plates.

The inside radius of curvature at the flange connecting the end to the cylindrical shell must not be less than four times the thickness of the end plate, and in no case less than 2·5 inches.

2. When the end has a manhole in it, $\frac{A}{32}$ inch must be added to the thickness of plate.

If the plate at the manhole is stiffened by flanging, the total depth of the flange from the outer surface, in inches, is to be at least

$$= \sqrt{t \times w},$$

t is the thickness of plate, in inches,
w is the minor axis of the hole, in inches.

3. When the end or crown is a complete hemisphere without stays or other supports, and is made in more than one plate, the working pressure to be allowed is given by the following formula :—

$$WP = \frac{(t-2)S \times J}{C \times R},$$

where t is the thickness of the plates, in 32nds of an inch ;
S is the minimum tensile strength of the steel plates, in tons per square inch ;
J is the strength of riveted joint per cent. of the solid plate ;
R is the inner radius of curvature, in inches ;
C is a coefficient which for treble riveting is 2·83, for double riveting is 2·9, and for single riveting is 3·3.

Thickness in 32nds Inch.	Metals.				Thickness in 32nds Inch.	Metals.			
	Wrought Iron.	Mild Steel.	Cast Iron.	Brass Muntz Metal.		Wrought Iron.	Mild Steel.	Cast Iron.	Brass Muntz Metal.
10	12·50	12·81	11·72	13·75	30	37·50	38·44	35·18	41·25
11	13·75	14·08	12·89	15·13	31	38·75	39·72	36·35	42·63
12	15·00	15·36	14·06	16·50	32	40·00	41·00	37·52	44·00
13	16·25	16·64	15·24	17·88	33	41·25	42·28	38·70	45·38
14	17·50	17·92	16·41	19·25	34	42·50	43·56	39·87	46·75
15	18·75	19·20	17·58	20·63	35	43·75	44·84	41·04	48·13
16	20·00	20·10	18·75	22·00	36	45·00	46·12	42·21	49·50
17	21·25	21·78	19·92	23·38	37	46·25	47·40	43·38	50·88
18	22·50	23·06	21·10	24·75	38	47·50	48·68	44·55	52·25
19	23·75	24·34	22·27	26·13	39	48·75	49·96	45·73	53·63
20	25·00	25·62	23·44	27·50	40	50·00	51·24	46·90	55·00
21	26·25	26·90	24·61	28·88	41	51·25	52·53	48·07	56·38
22	27·50	28·18	25·78	30·25	42	52·50	53·81	49·24	57·75
23	28·75	29·46	26·96	31·63	43	53·75	55·10	50·42	59·13
24	30·00	30·74	28·14	33·00	44	55·00	56·38	51·59	60·50
25	31·25	32·02	29·31	34·38	45	56·25	57·66	52·76	61·88
26	32·50	33·31	30·48	35·75	46	57·50	58·94	53·93	63·25
27	33·75	34·60	31·66	37·13	47	58·75	60·22	55·10	64·63
28	35·00	35·88	32·84	38·50	48	60·00	61·30	56·28	66·00
29	36·25	37·16	34·01	39·88	49	61·25	62·58	57·45	67·38

INDEX.

	PAGE
Acoustic conductivity of metals	506
Acting surface of screw propellers	207
Admiralty conditions, contracts for oil fuel	307, 308
— flexible wire ropes	431
— hydraulic tests	727
— instructions to boiler overseers	390
— oil specification	241
— rule for determining I.H.P., co-efficients for	45
— rules for steel forgings	446
— supervision of boilers	389
— tests for rivets	390
— — of boiler steel	340
— — of steam-pipes	285
— — of turbines	231
— — and trials of machinery details	727
Air pressure in stokeholds	315
— pump barrels	161
— — lever pins	175
— — pipes of	162
— — rods	160
— — valves	160
— — — size of	161
— pumps, capacity of	159
— — kinetic system	158
— — ordinary	158
— — position of	158
— — speeds of	161
— — Weir's dual	159
— required by liquid fuels	240
— — for combustion	301
Allowance for wetted skin	38
— of steam room in various boilers	336
— of total heating surface	385
Alloys, fusible, melting-point of	476
Aluminium, particulars of	451
Amethyst, trials of	235
Angle-bars and stays, Board of Trade tests	346
— — weight of	486
Angles of entrance of steamships, table of	37
Atmospheric air, properties of	300
Auxiliary machinery, consumpt of	320
— valves of cylinders	96
Balancing of engines	287
Ballast tank suctions, British Corporation rules for	637
Bar links or quadrants	187
— — proportions of	190
— — rules for	189
— list as rolled in Great Britain	459
— steel, Board of Trade rules for	346
Beams, bending moments on	463
— deflection of	466
— forms of, for uniform strength	469
Bearing brasses for small engines	147
Bearings, Michel design	128
— and pin, pressure on	128
Bevis-Gibson torsion meters	17
Bilge and ballast fittings, British Corporation rules	636
— and other ship's pipes	418
— pipes and fittings, Board of Trade requirements	429
— — British Corporation rules	635
— pumps	172
— — Board of Trade and Lloyd's rules for	173
— suction pipes	167
— — — sizes of	419
— suctions, British Corporation rules for	637
Blades of screw propellor	202
— of turbines	229
Blow-off and scum cocks, Bureau Veritas rules	644
— pipes, etc. for boilers	399
B.M.E.D. & C. Committee's rules for boiler stays	372
— — — for furnaces	374
— — — for girders	371
— — — for safety-valves	396–406
— — tests recommended	733
Board of Trade factors of safety	354
— — rules for angle and stay bars	346
— — — for bilge pumps, pipes, etc.	173
— — — for blow-off cocks, etc.	264
— — — for boiler mountings	401
— — — for boiler stays	362
— — — for copper pipes	278
— — — for distilling apparatus	417
— — — for evaporators	376
— — — for feed pumps, pipes, etc.	171
— — — for furnaces	374
— — — for motor boats	254
— — — for petrol engines	252
— — — for pipes	276

PAGE

Board of Trade rules for riveted joints 381
— — — for safety-valves . . . 402
— — — for sea fittings . . . 263
— — — for shafts 136
— — — for ship fittings . . . 422
— — — for steering gear . . . 687
— — — for turbine shafts . . 137
— — — miscellaneous . . . 422
— — tests of boiler steel . . . 341
— — — of steel 350
— — — of turbines 231
— — — required by 729
— — thickness of copper and steel pipes 279
Boiler construction, materials of . 329
— efficiency, affected by fire-bars . 309
— fittings, Bureau Veritas rules for 645
— gauges, German Government rules for 670
— installations, weight of . . 338
— materials, tests of 388
— mountings and fittings . . . 406
— — Board of Trade rules . . 401
— — Lloyd's rules for . . . 407, 742
— seats and attachments . . . 413
— shell joints, strength of . . 352
— — openings in 355
— — plates, differing tensiles . 354
— — — of various strengths . . 354
Boiler shells, cylindrical—
Board of Trade rules . . . 352
U.S.A. Government rules . . 667
Lloyd's Register . . . 359, 745
Boiler stays, U.S.A. Government rules for 668
— — Lloyd's Register . . . 372, 750
— — — rules for 372
— — pitch, etc., of Board of Trade rules 362, 370
— — rules for 371
— steel, Admiralty tests of . . 340
— — Board of Trade tests of . . 341
— — Bureau Veritas and tests of . 664
U.S.A. Government rules, and tests of 667
— — Lloyd's tests of 356
— tubes, Lloyd's rules for . . . 689
— — pitch of 337
— — prices, etc., of 490
— — size of 330
— — weight of 488
— work, Admiralty inspection . 389
Boilers, British Corporation rules . 621
— capabilities of various . . . 316
— circulation in 330
— comparison of weight of . . 339
— construction of 340
— — British Corporation rules . 629
— — Bureau Veritas rules . . 643
— cylindrical 323
— — particulars of 332

PAGE

Boilers, cylindrical, weight of . 332
— double-ended 323
— efficiency of 316, 326
— express, particulars of . . . 319
— flat plates of, thickness of . . 360
— furnaces of, rules for . . . 374
— gauges on 407
— German Government rules for . 667
— gunboat type 323
— heating surface of . . . 324, 325
— Hohenstein 324
— hydraulic testing of . . . 374
— in general 299
— kinds of marine 322
— modern cylindrical, scantling of 334
— riveting of 352
— small tube varieties . . . 324
— stays for, tests, etc., Board of Trade 370
— — for, tests of, Lloyd's . . . 372
— steam, room of 336
— stop valves of 395
— strength, U.S.A. Government rules for 667
— sundry hydraulic tests . . . 374
— tests of, Bureau Veritas rules . 642
— thickness of shell plates . . 351
— valves on 407
— various, trials of 317
— — types 324
— water spaces in 336
— — tube, kinds of . . . 322, 323
— — tube, rules for . . . 357, 670
— weight of cylindrical . . . 325
— working pressure, Lloyd's . . 359
— — — Board of Trade . . . 353
Boiling-point of sea-water . . 502
Bolts for cylinders and columns . 97
— for main bearings 145
— of screw blades . . . 211, 212
— of shaft couplings 131
Bombay, distance from other ports 722
Bosses of screw propellers . . 210
Brass plates, pipes, and turbine blades, Bureau Veritas rules . 663
— rods, sheets, etc. 694
— various kinds of 451
Brasses for connecting rods . . 119
— for main bearings 147
British Corporation rules, general 621
— — — for electric lighting . . 638
— — — for shafting 630
— — — for steel 447
— — tests for steel 622
Bronze, Admiralty, particulars of 449
— manganese, particulars of . . 449
— or gun-metal, strength, etc., of 448
— phosphor, particulars of . . 449
— various, by Bull's Metal Company 456
— — by Parsons' M.B. Company . 450
— — by Stone & Company . . 450

	PAGE
Bronze, various products of Delta Metal Company	452
Bronzes, zinc, particulars of	449
Built-up cranks, sizes of	132
Bunkers, fittings in, Bureau Veritas rules	643
Bureau Veritas rules, etc., general	641
— — — for electric lighting	664
— — — for shafting	654
— — — oil-engine shafts	245
— — steel for boilers	660
— — tests required by	729
Bushes for stern tubes	143
Butt joints of shells, Bureau Veritas rules	648
Cairncross, with geared turbines	237
Calcutta, distance from other ports	718
Calorific value of fuels	299
Cape Town, distance from other ports	718
Caps for main bearings	145, 146
Carbonic oxides, composition of	300
Cardiff, distance from other ports	707
Casings (brass) for screw shafts	136
Cast iron, Admiralty requirements	439
— — contraction of castings	438
— — mixtures of	438
— — strength of, etc.	437
— steel, particulars of	440
— — and bronze pipes	277
Castings, British Corporation rules	632
— steel	346
Centre of gravity of machinery	497
Centrifugal pump wheels	166
— pumps	165
— stresses on screw propellers	212
Chain cables, Lloyd's tests	430
— — proportions of	432
— — safe working load on	430
— — short-link, Admiralty tests	429
— — stud-link, Admiralty	429
Chains and ropes, Admiralty tests of	429
— weight of	432
Channel steamers, performance of	226
Chemical compositions	300
Circles, areas of	550
— circumferences of	557
— small, areas and circumference	564
Circulating apparatus for boilers	401
— pumps	157, 163
— — centrifugal	165
— — efficiency of	167
— — pipes of	165
— — sizes of	164
Circulation of water, apparatus for	401
— — in boilers	330
Classification, Bureau Veritas rules	641
Clearance, effect on mean pressure	70
— of pistons	99
Coal, anthracite	299
Coal, bituminous	299
— consumption per day	508
— kinds of, and value	299
— lignite	299
Co-efficient of fineness of ships	43
Column bolts	149
Columns of engines	148
Combustion, air required for	300
— and evaporation in various boilers	317
— — rates of	318
— chambers, Board of Trade rules for	369
— — girders on	369
— — tops	369
— — numbers of	323
— rates, of, in practice	308, 314
Compasses, Bureau Veritas rules	666
Compensating rings at manholes	353
Compensation rings of boilers,	355
Compound *versus* triple engines	57
Compression on plates	370
Condenser, form of	152
— tube alloys	451
— — ferrules	156
— — plates	156
— — — tubes per sq. ft.	157
— tubes	155, 693
— — and ferrules, standards	155
— — length	155
— — pitch of	157
— — weight of	494
Condensers, jet	149
— Morrison's	152
— surface, advantage of	151
— tests of, Bureau Veritas rules	642
— Weir's Uniflow	152
Conductivity of materials	477, 506
Cones, volume and surface of	566
Connecting-rod bolts	118
— — brasses	119–121
— — caps	118
— — ends	120
— — gudgeons	118
Connecting-rods, rules for	116
Consumption of fuel, etc., on various ships	321
— — per day	508
— of oil fuel per day	509
Contraflow condenser	152
Converting evaporation results to and from	337
Cooling surface	152, 153
— — amount of	153
— water, application of	157
— — flow of	154
— — pipes	166
— — quantity of	155
— — ratio to steam	155
Copper, Admiralty requirements	448
— alloy for pipes	447
— pipes	273–278

PAGE
Copper pipes, by Board of Trade rules 278, 369
— — flanges. and bolts for . . 275
— plates and pipes, Bureau Veritas rules 663
— strength, etc., of 447
— tubes, Lloyd's rules for . 696, 739
— — thickness of 272
— — weight of 492
— wire, strength of 447
Corrugated furnaces, standard sizes 373
Coupling bolts for shafts, particulars of 131, 738
— — sizes of 131
Covers of cylinders 86
Crank-arms 134
— keys, rules for 135
— pins, size of 131, 134
— shaft journals, size of . . . 131
— shafts, built up 132
— — in general 125
— — of paddle engines . . . 132
— — of screw engines . . . 128
— — loads on 129
— — size of arms 130
Crossed valve-rods 183
Crossheads for air-pumps, etc. . . 175
— of piston-rods 116, 118
Cube root of squares, tables of . 52
Cubes, squares, and roots . . . 567
Curves of speed, power, etc., of ships 39
Cylinder arrangements in compound systems 60
— columns 97
— cover joints 93
— covers 86
— — cast steel 91
— diameter of, for power . . . 79
— drain cocks 95
— ends 86, 88
— liners, thickness of, in cast iron 85
— — — in steel 86
— of oil engines, number of . . 242
— ports and passages 79
— false faces 98
— feet 97
— — bolts 97
— pass valves 96
— ratios 57
— relief valves 94, 95
— steam passages, thickness of . 90
— valve-boxes 89
Cylinders, general strength of . 82
— horizontal 98
— of oil engines, thickness of . . 251
— oscillating 98
— of reversing engines . . . 190
— size of 78
— sizes of, for N.H.P. . . . 5
— steam, thickness of . . . 82
— tests of, Bureau Veritas rules . 641

PAGE
Cylindrical boilers, scantlings of . 334
— — *v.* water-tube boilers . . 328

DELTA Metal Company, product of 452
Denny-Edgecombe torsion meter . 17
Depth of water for speed trials . 48
Destroyers, Yarrow's productions . 237
Diagrams (Zeuner's) for valve motions 181
Diameter of paddle-wheels . . 215
Diana, H.M.S., steam consumption of 320
Diesel engines, fittings, Lloyd's rules for 698
— — N.H.P., Lloyd's rules for . 695
— — shafts, Lloyd's rules for . 696
— — ships fitted with . . . 703
Dished ends, Bureau Veritas rules 652
Displacements with Index ⅄ R.D.S., Table XVIII. 52
Distance of ports from one another 706
Distilling apparatus, Board of Trade requirements . . . 428
Dover, distance from other ports . 718
Drain cocks of cylinders . . . 95-96
Draining of steam pipes . . . 279
Draught in funnels and chimneys . 309
Duralumin, particulars of . . 452

ECCENTRIC rods 194
— sheaves 191
— — and straps 193
— straps 191
Eccentrics, rules for 192
Economic speed, maximum . . 39
— — ship's length for . . . 39
Effective horse-power, to determine co-efficient for 46
Efficiency, as affected by steam jackets 27
— mechanical 17, 18
— of boilers 17, 18
— of engines, general . . . 19
— of marine machinery . . . 17
— of propellers 17
— of pumps 167
— of steam 17
— of steam-engines 56
— propulsive 17
— thermal 17
Electric light, etc., Lloyd's rules for 172, 672
— — in oil-carrying vessels . . 676
— lighting, British Corporation rules for 638
— — Bureau Veritas rules . . 664
Electrical conductivity of metals . 506
— resistance of metals . . . 478
Engine and boiler rooms, Lloyd's rules 414
— — seatings 411
— and boilers, weights of . . 496

	PAGE
Engine columns	148
— framings and foundations	148
— friction, Dewrance's experiments	21
— — Froude's method	20
— — Tower's experiments	21
— power measurement	2
— rules of N.E. coast standards	719
Engines, construction of—	
British Corporation rules	633
Bureau Veritas rules	643
Engines for reversing gears	199
— for turning gears	200, 201
— reciprocating	1
Equivalent evaporation from and at 212°	337
— — per lb. of fuel	331–337
— mean pressures	66
Escape valves on cylinders	94–95
Estimated horse-power	3
Evaporation, equivalent, from and at 212°	331–337
— — from and at various temperatures	525
Evaporative power of fuel	301
Evaporators, Board of Trade rules for	374
Exhaust pipes of steel	284
— steam, properties of	150
Expansion of metals by heat	474
— of steam isothermally	67
Express boilers, capabilities of	319
— — weights of	319
Extras charged by steel manufacturers	460
FACTORS of safety for boilers, Board of Trade	354
Feathering floats	216
Feed heaters	171
— pipes	169
— — diameter of	398
— — sizes of	170
— pump plungers	170
— pumps	168
— — capacity of	169
— — pipes, etc., Board of Trade rules for	171
— — — — Lloyd's rules for	172
— tanks	171
— valves for boilers	397
— water, gross	168
— — solid matter in	499
— — velocity of flow	170
Feet and metres, tables of	536
— of cylinders	97
Ferrules for condenser tubes	156
Fineness, co-efficients of, tables for	44
— of ships, rules for	43
Fire-bars, bridges, etc.	408
Fire hose and fittings, Board of Trade requirements	425
Flanges for steel pipes	285, 286
Flanges of copper pipes	275
— of screw blades	212
Flash-points of oil fuels	239
Flat bars, weight of	483
— plates of boilers	360
— — — stays of	360
Flat surfaces—	
Greatest fluid pressure on	471
Maximum pressure on various	471, 473
Of boiler, Board of Trade rules for	360
— — British Corporation rules for	631
— — Bureau Veritas rules for	650
— — Lloyd's Register rules for	366, 750
Of boilers	360, 366
Floats for paddle-wheel, number of	216
Forced draught	314
Forgings, Board of Trade rules for	347
— British Corporation rules	632
Föttinger torsion meter	16
Fourth power of numbers	612
— — of shaft diameters	613
— root of numbers	612
Framings of engines	148
Fresh-water distillers, Board of Trade rules for	427
Friction, co-efficients of	504
— of skins of ships	46, 528, 546
Frictional resistance	23
— — to total ratio	32
Fuel consumption of oil engines	246, 251
— for internal combustion engines	240
— liquid characteristics of	303
— of various kinds, composition	302
— weight of consumed	310
Fuels for oil engines	239
— various kinds	299
Funnels and casings, riveting	313
— — chimneys	309
— — — draught in	309
— capacity of, for fuel	312
— effect of, on fuel consumption	310
— height of	311
— scantlings of	313
— size of	310 311
Furnace fittings, etc.	408
Furnaces, corrugated, rules for	373
Furnaces of boilers—	
Board of Trade rules for	373
Furnaces of boilers, rules for	373
— size of	373
Fusible alloys, melting-point of	476
— plugs, in boilers	670
GAS engines for ships	239
Gauges on boilers	407
Geared turbines for propulsion	224
German Government rules—	
For boilers	667
For furnaces	667
Gibraltar, distance from other ports	706

PAGE

Girders for combustion chamber tops 369, 370
Glands and stuffing boxes . . 101
Glasgow, distance from other ports 708
Grate area, quantity of 333
Gratings, ladders, and platforms . 410
Gudgeons for connecting-rods . 118
Guides for piston-rods . . . 115
— — slide valve rods 187

HALIFAX, distance from other ports 717
Hamburg, distance from other ports 711
Hatchways, Lloyd's rules . . 416
Hawsers, steel wire, strengths of . 432
Heat on metals, effects of . . 474
— total of evaporation . . . 525
Heating surface, allowance of, in ships 335
— — of superheaters 261
— surfaces, condition of . . . 330
— — efficiency of 329
— — total, required 333
Hemispherical ends of cylinders . 365
Hollow shafts 124
— — weight of 482
Hong Kong, distance from other ports 717
Hopkinson-Thring torsion meter . 17
Horse-power, estimated . . . 3
— indicated 9
— net 10
— nominal 3
— of oil engines, rules . . . 250
— per 100 feet of wetted skin for various speeds 38
— propeller 10
— shaft 9, 15
— thrust 10
— to drive a ship, rules for 28, 29, 37, 38
— tow rope 10
— turbines 232
— various rules for . . . 4, 5, 6
Howden's forced draught . . . 314
Hull, distance from other ports . 709
Hydraulic tests of boilers . . 376, 670
— — Admiralty 727
— — Italian Government . . . 728
Hyperbolic logarithms . . . 614

INCHES and millimetres, tables of 527
India-rubber valves, Admiralty specification for 161
Indicated horse-power . . . 9
— — multipliers for 12
— — of oil engines 250
Indicator diagrams 11
Inertia, effect of, on engine parts . 127
— forces on engines 287
Injection orifice, size of . . . 151
— water, quantity of 149
Inlet valves 222
Insulating materials for boilers, etc. 670

PAGE

Internal combustion engines . . 239
— — — Lloyd's rules 256
— steam pipes of boilers . . . 395
Iron bars, strength, etc., of . . 439
— — weight of 480
— cast, particulars of . . . 437
— pipes, Board of Trade thickness of 274
— stay bars, Board of Trade allowance 371
Italian Government tests and trials of machinery details . . . 728
— — hydraulic tests 728

JET condensers 149
Joy's valve gear 194

KEYS for propellers 136
Kilogrammes per sq. cm. to lbs. per sq. inch 545
Kilometres and Admiralty knots 530
— and miles, standard and nautical 527
Kirk's analysis of steamship trials 36
Knots, miles, kilometres, tables of 527

LADDERS, gratings, etc. . . 410
Lead of slide valves 177
Lead pipes, weight of 495
Leith, distance from other ports . 714
Levers for air pumps 174
— for reversing links 189
Link motion, drag rods . . . 189
— — effect of crossed rods . . 183
— — — of obliquity of eccentric rods 183, 184
— — levers 189
— — notched up 182
— — open rods 183
Links, double bar 191
— for pumps and levers . . . 175
Liquid fuels in practice . . . 305
— — particulars of 303
Liverpool, distance from other ports 711
Lloyd's Register—
Rules for bilge and other pipes . 418
— — — pumps, etc. . 173, 417, 732
— — boiler mountings . . 407, 743
— — — rooms 414
— — — stays 371
— — — tubes 693
— — boilers . . . 356-359, 740
— — brass rods, sheets, etc. . 694
— — carrying liquid fuel . . 307
— — condenser tubes, etc. . . 693
— — copper tubes . . . 696, 739
— — Diesel engine fittings . . 698
— — — — shafts 696
— — electric light . . . 172, 672
— — engine rooms 414
— — feed pumps, pipes, etc. 172, 731
— — flat surfaces . . . 368, 750
— — furnaces 375, 748

PAGE

Lloyd's Register—
Rules for internal combustion engines 256
— — list of spare gear . . . 421
— — motor boats 256
— — nominal horse-power . . 6
— — — — of Diesel engines . 699
— — oil-engine shafts . . . 244
— — pipes 269, 734
— — refrigerating machinery . 682
— — screw shafts . . 705, 735
— — sea fittings . . . 265, 733
— — shafts 138, 734
— — spare gear for refrigerating machinery . . . 683
— — steam pipes . . . 269, 739
— — steel boilers 379
— — — forgings . . . 441, 447
— — steering gear 686
— — survey of boilers . . . 421
— — — of machinery . . . 419
— — surveys of refrigerating machinery . . . 683
— — tests of boiler steel . . 356
— — — of steel and other materials 688
Logarithms, hyperbolic . . . 614
London, distance from other ports 718
Low-pressure steam, particulars of 150
Lubricating oils, characteristics of 503
Lusitania, S.S., steam consumption of 320
— — trials of 233

MAIN bearing bolts and caps . . 145
— — brasses 147
— bearings of engines 145
Malleable cast iron, Board of Trade requirements 144
Manganese bronze, particulars of . 449
Manholes, compensating rings . 353
Marine engines, trials of, Table XXV. 74
Materials, weight of 479
Mean pressure, equivalent of a compound system 66
— — of dry steam expanding adiabatically, Table XXII. 71
— — of steam expanding isothermically 67
— — of steam as found in practice 73
— — referred, of a compound system 66
— pressures, allowing for clearance 70
— speed of ships, rules for . 47, 48
Measurement of engine power . 2
Measures, English and metrical . 526
Mechanical efficiency 17
— equivalent of heat 300
Melting-point of various metals . 476

PAGE

Melting-points of metals (Prof. Carnelly's Constants) . . . 734
Metallic packings, Table XLII. . 102
Metals, conductivity of . . . 477
— electrical resistance of . . . 489
— melting-point of . . . 476, 730
— specific heat of 477
— strength of, when heated . . 475
— various, effect of temperature on 475
— — expansion of, by heat . . 474
— various prices of 456
— — properties of 454
Meters, to ascertain torsion . . 16
Metres and feet, table of . . . 536
Metrical measures, standards of . 526
Michel's Thrust blocks 141
Mild steel bars, weight of . . . 481
Miles, statute and nautical, and kilometres 527
Millimetres and inches, tables of . 531
Minimum length of ships for speed 36
Model experiments for ships . . 42
Modulus of certain sections . . 467
Moments of inertia of sections . 467
Monel metal, particulars of . . 452
Motor boats, Board of Trade rules for 252
Mountings, boiler, Board of Trade rules for 401
Multipliers for I.H.P. 12

NAVAL brass, particulars of . . 451
New York, distance from other ports 714
Nomenclature of boilers . . . 617
— of shafts 617
Nominal horse-power 3
— — — foreign countries . . . 9
— — — Lloyd's rule 6
Normania, S.S., efficiency of . . 223
— — trials of 235
North-East Coast standard engines 719
Notching up slide valve link motion, effect of 182
Number of screw blades . . . 202
— of propellers 201
Nuts and bolts, dimensions of . 105

OIL, Admiralty requirements . . 241
— and liquid fuel, air necessary for 240
— — — composition of . . . 240
— — — flash and boiling-points of 239
— — — specific gravity of . . . 240
— — — thermal value of . . . 240
— consumption per day . . . 509
— creosote, or tar 241
— engines, Bureau Veritas, shafts . 245
— — characteristics of . . . 242
— — classified 241
— — crank shafts 243
— — cycles of 241
— — cylinders of 242, 251

	PAGE
nes, design of	242
iciency of	247
ships	239
el consumption	246, 251
ide shoes of	243
licated H.P. of	250
an pressure in	248
tons of	242
mps for	246
ersing of	242
olutions of	248
ace occupied by	249
ts of	247
ious	1
ight of	249
Admiralty condition of tract	308
ested in U.S.A. navy	306
sumption of	304
ious kinds of	303
osity of	306
	241
halt	241
g and flash points of	504
rnian	241
n	241
s, criterion for tests	503
ty of	502
in decks, Lloyd's rules	415
ng cylinders	98
er pistons	185
ials of	234
metallic	102
ids	100, 101
ist blocks	142
gine cranks	132, 133
ings	412
s, rules for size of shafts	124
immersion of	217
, rules for	220
particulars of	133
propeller	215
, area of	216
gn of	218, 219
ieter of	215
ensions of	220
ering floats	216
ies of	217
ber of floats	216
al	215
of	216
ness of floats	217
steam through pipes	261, 393
ls	299
stons, diagram of	180
ines, Board of Trade rules	252
bronze, particulars of	449
mps	175
gements	279
pipe arrangements	269
nd T-pieces	276

	PAGE
Pipes bilge	167
— blow-off and scum	272
— bronze and cast steel	277
— copper, Board of Trade thickness	278
— — brazed	276
— — flanges of	275
— — thickness of	273
— discharge	272
— exhaust, thickness, etc.	284
— expansion joints, Board of Trade	279
— for cooling water	166
— for steam, draining off	279
— for steam, steel	280
— Lloyd's rules for	279
— miscellaneous	276
— of air pumps, size of	162
— of circulating pumps	165
— solid drawn steel	281
— — — — tests for	281
— steam and feed	269
tests of, Bureau Veritas	642
— steel, bending of	285
— — steam	280, 283
— — welded, etc.	274
— test of Board of Trade	278
Piston clearance	99
— path, diagram of	180
— — of an oscillating cylinder	185
Piston-rod crossheads	116, 118
— — guides	115
Piston-rods, construction of	114, 115
— — loads on	113
— — rules for size	114
— slide valves, rods of	187
— speed	76
Pistons, cast iron	109, 110
— — steel	106, 107
— forged steel	109
— general construction	112
— rules for	106, 109
— stroke of	78
Pitch of rivets, maximum	354
— of screw propellers	206
— ratio of screws	204
Plate list as rolled in Great Britain	458
Plates subject to compression— Board of Trade rules	370
Lloyd's rules for	370
Ports and passages of cylinder	79, 81
Possible work from 1 lb. of steam	28
Power to speed, relations of	30, 31, 37, 48
— transmitted by shafts	616
Pressure of water due to head	507
— on pistons, effective	13
Pressures on bearings and pins when working	128
— on thrust blocks	139
— per sq. inch and kilos. per sq. centimetre	544
Prices of various metals and alloys	456
Prime movers on shipboard	1

	PAGE
Prism co-efficients, table of, for various ships	44
Prismatic co-efficients	39
Propeller blades, shape of, section of	203
— driven by turbines, reversal of	223
— shafts	135
Propulsion of ships	27
Propulsive efficiency	17
Pump crossheads	175
— levers, links, etc.	174
Pumping arrangements, British Corporation rules	636
Pumps, air and circulating, scantlings of	163
— barrels of	161
— circulating water	163
— feed	168
— for bilge water	172
— for ship's use—	
Board of Trade rules	422
Lloyd's rules	417
— relief valves for	169
Pure water, weight of hot	400
QUADRUPLE compound engine, horse-power of, Table III.	8
— compound engines, trials of	268
— screw cruiser trials	236
RATIOS of cylinders in compound engines	57–59
Receiver safety valve	95
Reciprocating engines	1
Refractory materials for furnaces	620
Refrigerating machinery, Lloyd's rules	678
Relief rings on valves	177
— valves for pumps	169
Reserve feed water-tanks	171
Resistance due to skin friction of ships	28, 46
— of ships	27
— — due to eddies	28
— — — to waves	28
— of various surfaces to water passage	47
— residual, in ships	28
Reversing gear	197
— — cylinders	198
— — engines	199
— — weigh-shafts	199
— quadrants	188
Revolution, rate of, in marine engines	77
Revolutions of oil engines	248
— of turbines	226
Riveted joints, Board of Trade rules	380
— — double and quadruple special	387
— — examples of various	384
— — maximum pitch of rivets	383
Riveted joints, quadruple examples	386
— — relative strengths of	389
— — treble examples	385
Riveting, multiple, pitches of	354
— of longitudinal seams	353
Rivets, Board of Trade tests	347
— maximum pitch of	354
Rods for slide valves	188
— of circulating pumps	165
Roots, square and cube	567
Ropes and cordage, tarred hemp	436
— hemp, particulars of	433
— steel wire flexible	431–433
— tarred hemp, Admiralty	436
— wire, Bullivant's	434
Round bars and shafts as rolled	459–461
Rule for girders, B.M.E.D. & C. Committee's	371
Rules for boiler stays	371
— for N.E. coast standard engines	719
— for shafting, Bureau Veritas	654
Russian weights and measures compared	725
SAFETY valves, area of	395
— — Admiralty requirements	395–405
— — B.M.E.D. & C. Committee Rules for	396, 406
— — Board of Trade rules	395, 402
— — Bureau Veritas rules	644
— — French rules	396
— — U.S.A. Government rules	667
— — on cylinder receiver	95
— — springs for	397
— — tests of	407
Screw engines, rules for size of shafts	124
— propeller, numbers of	202
— — acting surface	207
— — blades, number of	203
— — — sections	204
— — bolts of blades	211
— — bosses	212
— — centrifugal stress	212
— — diameter of	203, 205
— — dimensions of	203
— — flanges	212
— — for turbines	227
— — keys	136
— — materials of	203
— — particulars of	213
— — pitch of	206
— — pitch ratios	204
— — slip of	204
— — surface of	204
— — thickness of	207
— — thrust of	204
— — weight of	211
— shaft casings	136
— shafts	135
— — Lloyd's rules for	705
— threads, particulars of	104
— — per inch of boiler stays	374

	PAGE
Screw threads, Whitworth gas	494
Screws, number of	201
Sea and other ship valves, Lloyd's rules	415
— fittings, Lloyd's rules for	264–265
— valves generally	222
— water, boiling-points of	502
— — in various parts of world	500
— — supply to engine room	222
— — temperature of	153
— waters, solids in	500
— — weight of, cub. ft.	501
Seaports, distance of, from one another	710–718
Seatings for engines and boilers	411
Seaton's rules for nominal horse-power	6
Sections of screw blades	203
Segments of circles, areas of	547
Semi-Diesel engines	252
Shaft couplings	131
— diameters, fourth power of	615
— horse-power	9–15
— of paddle engines	124
— tunnels, Lloyd's rules	414
Shafting, British Corporation rules	630
— Board of Trade rules for	136, 137
— Bureau Veritas rules for	658
— Lloyd's rules for	138, 734
— paddle, Bureau Veritas rules	657
— torque on	122
Shafts, fourth power of diameters	613
— nomenclature of	617
— of paddle engines	124
— of screw engines	124
— of turbines	228
— power transmitted by	616
— size of, to resist torsion	123
— tail, crank, etc.	137
— weight of hollow	482
— — of steel	482
Shape of screw blades	203
Sheet metals, weight of	495
Shell plates of boilers	354
Ships, fineness of	43
— fitted with Diesel engines	703
— model, experiments with	42
— power and curves of	39, 40
— propulsion of	27
— resistance of	27
— skin friction of	28, 46
Size of cylinders corresponding to N.H.P.	5
— — to calculate	78
Skin fittings of steamers	262
Slide valve proportions	177–179
— — rods, sizes of	187
— — section of	178
— valves, effect of notching or linking up, diagram	182
— — guides for rods	187
— — lead of, for steam	177
— — loads on rods	187
Slide valves period of release	177
— — relief rings on	177
— — springs on back of	177
— — travel and surface of	176
— — Zeuner's diagram of motion	180
Slip, apparent, of paddle-wheels	216
— — of screws	204
— ratio of turbine screws	227
— real, of screws	205
Slot links for reversing	187
— — rules for	188
Solid-drawn steel pipes	285
Solid matter in sea-waters	500
Sound, conductivity of, by metals	506
Sounding pipes, Board of Trade requirements	423
Southampton, distance from other ports	715
Spare gear, Board of Trade requirements	426
— — British Corporation rules	635
— — Bureau Veritas rules	659
— — Lloyd's requirements	421
Specific heat, definition of	301
— — of metals	476
— — of superheated steam	259
— — of various materials	477
Specification, standard, N.E. coast, triples	719
Speed and power curves, of ships	39
— of air pumps	161
— of pistons	76
— of ship and power, relation of	43
— of ship in knots from measured mile observations, Table XVII.	49
— of ships, economic maximum	39
— of steam through ports and pipes	80
— true mean, rules for	47
Spheres, volume and surface of	565
Springs for safety valves	397
— on back of slide valves	177
Square feet and square metres	540
— metres and square feet	541
Squares, cubes, and roots, etc.	567
Standard specification for triple compound engines, N.E. coast	719
Stay for flat surfaces of boilers	362
— tubes in boilers	370
Staybars of iron, Board of Trade allowance	371
— surfaces sustained by	372, 373
— threads of standard	374
Staying of boilers—	
Board of Trade rules for	370
British Corporation rules for	627
Bureau Veritas rules for	651
U.S.A. Government rules for	667
Lloyd's Register rules for	371
Stays for flat plates	360
Steam and feed pipes	269, 635
— and fuel consumption of various steamers	321

PAGE

Steam, consumption 14
— — H.M.S. *Amethyst* . . . 235
— — how affected by vacuum . . 152
— — of ships 320
— — of turbines 232
— — S.S. *Lusitania* 233
— — S.S. *Normania* 235
— — S.S. *Otaki* 234
— — trial, S.S. *Cairncross* . . 238
— cylinders, thickness . . 82, 83, 84
— efficiencies of engines . . . 56
— expanding adiabatically . . 68
— — factors for mean pressures of, Table XXI. 69
— — factors for mean pressures of, when with clearance, Table XXII. 70
— — isothermally 67
— flow of, through ports and passages . . . 80, 261, 393
— jackets, effect on efficiency . 27
— maximum output from one pound 23
— — work from one pound expanding 67
— of low pressure, characteristics of 150
— passing through pipes, weight of 393, 394
— pipes, Lloyd's rules for . . . 269
— — and valves, diameter of . . 392
— — of boilers, size of . . . 392
— port openings 177
— properties of 517
— room in various boilers . . . 336
— ships, trials of 33, 34, 36
— superheated 258
Steel bar sand plates, strength, etc., of 444
— — weight of 481, 484
— boilers, details of 351
— castings, Admiralty requirements 441
— — and forgings, test pieces for Board of Trade . . . 343
— — Board of Trade requirements 443
— — — — tests 348
— — comparative requirements . 445
— — cylinder covers 91
— — exhaust pipes, thickness of . 281
— — for boilers, Bureau Veritas rules 660
— — Lloyd's requirements . . 441
— — strength of, etc. 440
— — tests of, by various bodies . 445
— for boilers, Bureau Veritas . . 660
— forgings, Admiralty requirements 446
— — and castings, British Corporation rules 632
— — Board of Trade tests . . 347
— — Bureau Veritas rules . . 662
— — Lloyd's requirements . . 446

PAGE

Steel plates, angles and bars, as made 458
— — etc., British Corporation requirements . . . 447
— — — makers' extras for . . 460
— materials, extra for size, etc. . 460
— pistons 106, 109
— slabs, weight of 485
— steam pipes 280, 283
— — — thickness of . . . 280-282
— test pieces for Board of Trade . 342
— tests, British Corporation . . 622
— welding of 441
Steering engines, Board of Trade requirements 425
Steering gear—
Board of Trade requirements . 687
Lloyd's rules for 686
Stern bushes 143, 144
— tube fittings at outer end . . 136
— tubes 142
Stern tubes and bushes, sizes of, Table LV. 144
Stop valves 286
— — of boilers, size of . . . 395
Stowage of cargoes 566
Straps for eccentricities . . 191, 192
Strength of materials 436
Stresses on materials, safe limits of 462
Stroke of pistons 78
Studs and bolts, loads on . . . 103
— — stresses of 102
Stuffing-box of stern tubes . . 144
— boxes, sizes of, Table XLI. . . 100
Sunderland, distance from other ports 706
Superheated steam 258
— — maximum temperature . . 259
— — maximum work by . . . 261
— — specific heat of 259
— — total heat of 260
— — transmission of 260
Superheaters, heating surface . 261
Supervision of boiler work, Admiralty 389
Surface condensation 151
— condenser, capacity of . . . 157
— of plate, etc., B.M.E.D. & C. Committee rules, Table CXII. 373
— — — — for large stays, Table CXIII. 373
— of thrust collars . . . 139, 142
— of tubes of various sizes . . 498
Surfaces sustained by stay bars . 372
— — — — large 373
Surveys of machinery—
British Corporation rules . . 639
Bureau Veritas rules . . . 642
Lloyd's 419
Suspension pins for reversing links, position of 189
— of quadrants 190

	PAGE
TAIL shafts	135
Tanks for feed water	171
— reserve	171
Temperature effects on metals	475
— of sea-water in various parts of the world	153
Testing of boilers by water	376
Test pieces for Board of Trade	342
Tests, hydraulic for boilers	374
— for boiler materials	388
— of boiler plates, etc.	388
— of cylinders and turbines, Bureau Veritas rules	641
— of india-rubber (Admiralty)	161
— of solid-drawn steel pipes	285
— of welded and riveted steel pipes	285
— required by Board of Trade	729
— — by Bureau Veritas	729
Thermal conductivity of metals	477, 506
— efficiency	17
— — maximum of steam engines	18, 19
— units, British	300
Thermometers, comparison of	510
Thickness of flat plates—	
Board of Trade	360
Lloyd's	366, 368
Thickness of screw propellers	207, 208
Threads for gas pipes	494
Thrust block seatings	412
— blocks	139, 141
— — securing to the ship	142
— collars	140, 141
— normal mean	139
— of screw propellers	205
— shafts	139
Tiller, spare, Board of Trade requirements	424
Tools, etc., for distilling apparatus, Board of Trade requirements	427
Torpedo-boat destroyers	32
Torsion meters	16
— — for power of turbines	233
— on shafts	122
Torsional stiffness of shafts	122
Total heat of combustion	301
— — of evaporation	525
— — of superheated steam	260
Travel of slide valves	176
Trials of engines	74, 75
— — — four-crank triples	267
— — — quadruple compound	268
— — — three-crank triples	266
— of quadruple screw cruisers	236
— of steamships	33, 34
— of S.S. *Cairncross*	237
— of S.S. *Lusitania's* turbines	233
— of S.S. *Reina Victoria Eugenia*	337
Trick slide valve	178
Triple compound engines—	
Advantages of	57
N.E. coast specification of	719
Triple compound engines—	
Nominal horse-power of, Table II.	7
Triple compound engines—	
Trials of	266
Trunnions of oscillating cylinders, sizes of	99
Tube plates, boiler	360-387, 627
— — compression on	369
— — of tank boilers	360
— — of water tube boilers	357
— — wide surfaces of	369
— surface in boilers, efficiency of	155
Tubes, Board of Trade tests	348
— copper, weight of	492
— for condensers, composition of	155
— for water tube boilers	357
— in boilers, size of	330
— iron and steel, standard list of	490
— large steel, weight of	491
— of surface condenser, pitch of	157
— of tank boilers	373
— — — per square foot	374
— solid-drawn steel	281, 349
— surfaces of	374, 498
Turbine castings, Bureau Veritas rules	236
— — trials of	236
— propellers, slip ratio	227
— shafting, rules for	228
— — Bureau Veritas rules	658
Turbines, Admiralty tests of	231
— advantages of	222
— and reciprocators combined	2
— astern going	223
— blades of	229
— Board of Trade tests of	231
— combined with reciprocators	224
— designs of	229
— dimensions of	230
— division of, in ships	224
— efficiency of	223
— geared, etc.	2
— — to screw shafts	224
— Italian tests of	231
— kinds of	223
— rates of revolution	226
— rotor drums	229
— screw propeller for	227
— shaft horse-power of	232
— steam	222
— — consumption of, Rateau's rule	232
— torque	232
— torsion meters for	233
— various arrangements of	223
— *versus* reciprocators	224, 225
— weight of	231
Turning gears	199
— — wheels, worms, etc.	202
Twisting moment, equivalent	125
— moments, curves of	126

	PAGE
UNIFLOW condenser	152
United States, America, rules of—	
for boilers	667
for furnaces	668
for seamless tubes	669
for stays in boilers	669
for fusible plugs	670
VACUUM augmenter, Parsons'	160
— effect of, on steam consumption	152
— highest possible	155
Valve-boxes for feed-pumps	169
— casings of cylinders	90
— gear, general motion of	186
— — Joy's	194, 196
— rods, guides, etc.	188
Valves of air pump	160
— for admission of water	262
— for discharge of water	262
— on boilers	497
— rubber, Admiralty specification for	161
— steam cylinders, travel of	176
— stop and regulating	286
— to admit sea-water	222
Viscosity of various oils	502
WATER consumption of boilers	320
— delivered through pipes	614
— evaporated per lb. of fuel	331
— for jet condensing	149
— fresh, bulk and weight of	499
— gauges and test cock	401
— — Bureau Veritas rules	644
— — rules for	400
— level in boilers, Bureau Veritas rules	644
— pressure of, due to "head"	507
— salt, bulk and weight of	499
— spaces in boilers	336
— tube boilers, rule for	357–670
— weight of, at various temperatures	480
Water-feed, solid matter from	499
Weigh-shafts for reversing gear	198
Weight of angle-bars	486
— of boiler tubes	488
— of boilers, rules for	325
— of brass condenser tubes	494
— of copper tubes	492
— of cylindrical boilers	325, 332
— of engines and boilers	496
— of flat bars	483
— of hollow shafts	482
— of iron	438, 440
Weight of iron and steel bars	480
— of large steel tubes	491
— of lead pipes	495
— of marine installations	496
— of oil engines	249
— of pure water at various temperatures	400
— of screw propellers	211
— of sheet metals	495
— of steam passed through pipes	393, 394
— of steel shafts	482
Weights and measures, Russian, compared	725
— English and metrical	542
— of turbine installations	231
— of various boiler installations	338
— of various boilers compared	339
— of various materials	479
— of various sea-waters	301
— of water	498
Welding of steel	441
Wetted skin, to estimate—	
Kirk's rule	36
Mumford's rule	38
Seaton's rule	38
Wheels for turning engines	200
White metal in brasses	147
— metals, particulars of	453
Wire gauge, legal standard and metric do.	271
— gauges and their equivalents	270
— ropes and hawsers	431
Wood fuel, value of	299
Work from one pound of steam expanding and exhausting to atmosphere	25
Work from one pound of steam expanding and exhausting to condenser	24
Work from one pound of steam expanding and exhausting at pressure p_2	26
Working pressures on bearings and pins, rules for	128
— stresses on various metals	462
Worms for turning wheels	201
Wrought-iron bars, weight of	480, 483
— — pipes, Board of Trade rules	274
— — qualities of, etc.	439
YOKOHAMA, distance from other ports	716
ZEUNER'S diagram for slide valves	180

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